Report No. CG-D-03-89

FULL-SCALE TRIALS OF PRE-SWIRL VANES AND MODIFIED PROPELLERS ON A 41 FT. UTILITY BOAT

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FINAL REPORT
AUGUST 1988

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Prepared for:

U.S. Department Of Transportation
United States Coast Guard
Office of Engineering and Development
Washington, DC 20593-0001

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Technical Report Documentation Page

1. Report No.	2. Government Accession No.	3. Recipient's Catalog No.
CG-D-03-89		

4. Title and Subtitle **FULL-SCALE TRIALS OF PRE-SWIRL VANES AND MODIFIED PROPELLERS ON A 41 FT. UTILITY BOAT**	5. Report Date AUGUST 1988 6. Performing Organization Code
7. Author(s) Robert D. Sedat, Edward S. Purcell, and Carol L. Hervey	8. Performing Organization Report No. R&DC 13/88
9. Performing Organization Name and Address U.S. Coast Guard Research and Development Center Avery Point Groton, Connecticut 06340-6096	10. Work Unit No. (TRAIS) 11. Contract or Grant No.
12. Sponsoring Agency Name and Address Department of Transportation U.S. Coast Guard Office of Engineering and Development Washington, D.C. 20593-0001	13. Type of Report and Period Covered FINAL 14. Sponsoring Agency Code

15. Supplementary Notes

16. Abstract

The rotational energy which is normally lost in the slipstream of a propeller can be reduced by properly designed pre-swirl vanes, thus improving the propulsive efficiency. More recently, vanes have been designed which also reduce the circumferential wake variations in the inflow to the propeller, thus reducing cavitation and vibration. In these experiments, an asymmetric vane set, designed to reduce the circumferential wake variations caused by an inclined propeller shaft, is tested on a 41 ft. USCG twin-screw utility boat (UTB). Fuel consumption, shaft torque, RPM and panel accelerations above the propellers are measured for two different propeller designs, with and without vane sets. Without the vanes, the new propellers show fuel savings (compared to the original non-vaned propellers) which range from 1-7%, depending on boat speed. When the vanes are added to the new propellers, the fuel savings (compared to the original non-vaned propellers) increase to between 5-10%, depending on boat speed. Thus, significant reductions in fuel consumption are achieved, and even better results may be possible with more optimal design procedures.

17. Key Words	18. Distribution Statement
pre-swirl, vanes, propellers, asymmetric, fuel saving, cavitation	Document is available to the U.S. public through the National Technical Information Service, Springfield, Virginia 22161

19. Security Classif. (of this report)	20. SECURITY CLASSIF. (of this page)	21. No. of Pages	22. Price
UNCLASSIFIED	UNCLASSIFIED		

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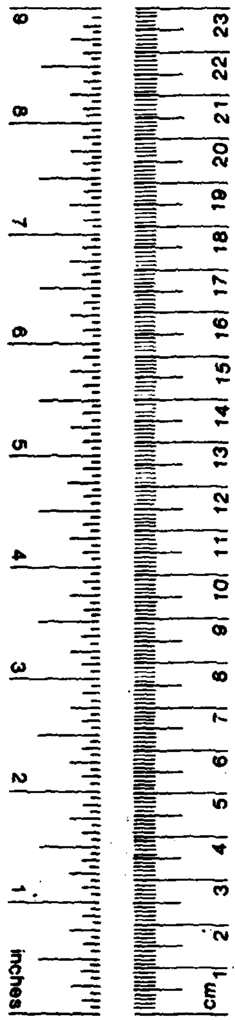
METRIC CONVERSION FACTORS

Approximate Conversions to Metric Measures

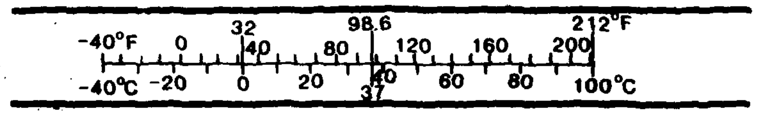
Symbol	When You Know	Multiply By	To Find	Symbol
		LENGTH		
in	inches	*2.5	centimeters	cm
ft	feet	30	centimeters	cm
yd	yards	0.9	meters	m
mi	miles	1.6	kilometers	km
		AREA		
in^2	square inches	6.5	square centimeters	cm^2
ft^2	square feet	0.09	square meters	m^2
yd^2	square yards	0.8	square meters	m^2
mi^2	square miles	2.6	square kilometers	km^2
	acres	0.4	hectares	ha
		MASS (WEIGHT)		
oz	ounces	28	grams	g
lb	pounds	0.45	kilograms	kg
	short tons (2000 lb)	0.9	tonnes	t
		VOLUME		
tsp	teaspoons	5	milliliters	ml
tbsp	tablespoons	15	milliliters	ml
fl oz	fluid ounces	30	milliliters	ml
c	cups	0.24	liters	l
pt	pints	0.47	liters	l
qt	quarts	0.95	liters	l
gal	gallons	3.8	liters	l
ft^3	cubic feet	0.03	cubic meters	m^3
yd^3	cubic yards	0.76	cubic meters	m^3
		TEMPERATURE (EXACT)		
°F	Fahrenheit temperature	5/9 (after subtracting 32)	Celsius temperature	°C

*1 in = 2.54 (exactly). For other exact conversions and more detailed tables, see NBS Misc. Publ. 286, Units of Weights and Measures. Price $2.25. SD Catalog No. C13.10.286.



Approximate Conversions from Metric Measures

Symbol	When You Know	Multiply By	To Find	Symbol
		LENGTH		
mm	millimeters	0.04	inches	in
cm	centimeters	0.4	inches	in
m	meters	3.3	feet	ft
m	meters	1.1	yards	yd
km	kilometers	0.6	miles	mi
		AREA		
cm^2	square centimeters	0.16	square inches	in^2
m^2	square meters	1.2	square yards	yd^2
km^2	square kilometers	0.4	square miles	mi^2
ha	hectares (10,000 m^2)	2.5	acres	
		MASS (WEIGHT)		
g	grams	0.035	ounces	oz
kg	kilograms	2.2	pounds	lb
t	tonnes (1000 kg)	1.1	short tons	
		VOLUME		
ml	milliliters	0.03	fluid ounces	fl oz
l	liters	0.125	cups	c
l	liters	2.1	pints	pt
l	liters	1.06	quarts	qt
l	liters	0.26	gallons	gal
m^3	cubic meters	35	cubic feet	ft^3
m^3	cubic meters	1.3	cubic yards	yd^3
		TEMPERATURE (EXACT)		
°C	Celsius temperature	9/5 (then add 32)	Fahrenheit temperature	°F



ACKNOWLEDGMENTS

We would like to thank all those who have made this work possible. The application of asymmetrical vanes for inclined flow was pioneered by Gary Larimer, formerly with the Coast Guard R&D group in Washington, D.C., and by Walter Gearhart of Penn State, who designed the current configuration. Thanks also to the able personnel of Coast Guard Group Long Island Sound and Station New Haven where these tests were conducted, particularly to CWO3 Mike Emch, BM1 Jack Godfrey, and MK1 Bob Jeffery.

TABLE OF CONTENTS

TABLE OF CONTENTS (Cont'd)

LIST OF ILLUSTRATIONS

LIST OF ILLUSTRATIONS (cont'd)

LIST OF TABLES

NOMENCLATURE

$C_t = T/(\rho A V_a^2)$	propeller loading coefficient
T	thrust
$V_a = V(1-w_x)$	speed of advance
A	propeller disk area
r	propeller radius
$J = V_a/nD$	advance ratio
$K_t = T/\rho n^2 D^4$	thrust coefficient
$K_Q = Q/\rho n^2 D^5$	torque coefficient
n	propeller revolutions/second
D	propeller diameter
w_x	axial wake fraction
w_t	tangential wake fraction
Q	torque
θp	propeller blade angle
β_i	hydrodynamic inflow angle
ϕ	section pitch angle
α	section angle of attack
α_i	"ideal" angle of attack
ρ	mass density of water
$1+\bar{a}$	axial induction factor at propeller disk
$1-\acute{a}$	tangential (swirl) induction factor at propeller disk
p	pressure

1.0 INTRODUCTION

It has long been known that rotational kinetic energy lost in the slipstream of the propeller reduces overall propulsive efficiency. Larimer [1] describes half a dozen patents filed between 1905 and 1951 for various symmetrical vane sets to reduce this lost energy. A typical configuration is shown in Figure 1-1. An early exposition of the theory behind pre-swirl vanes is given by Glauert [2], who predicts modest efficiency gains of 2-4 percent. There have been few test installations of such symmetric vane sets, however, and the use of pre-swirl vanes remains the exception rather than the rule.

In recent years, it has been proposed that asymmetric vane sets could also be used to modify the inflow to the propeller. This would enable the propeller blade sections to operate at a more nearly constant angle of attack during each revolution, and thus be more resistant to cavitation. This is a very intriguing possibility, particularly for vessels with severe cavitation problems, or for Naval missions where noise control is essential.

This paper will present the theory and design considerations for symmetric and asymmetric pre-swirl vane sets. The history of the development of an asymmetric vane set for the Coast Guard 41' utility boat (UTB) will be discussed. Published full-scale trial results will be reviewed, and additional data from a recent set of trials will be presented. Finally, a cost/benefit analysis is provided, and recommendations are made for future development of pre-swirl vane technology.

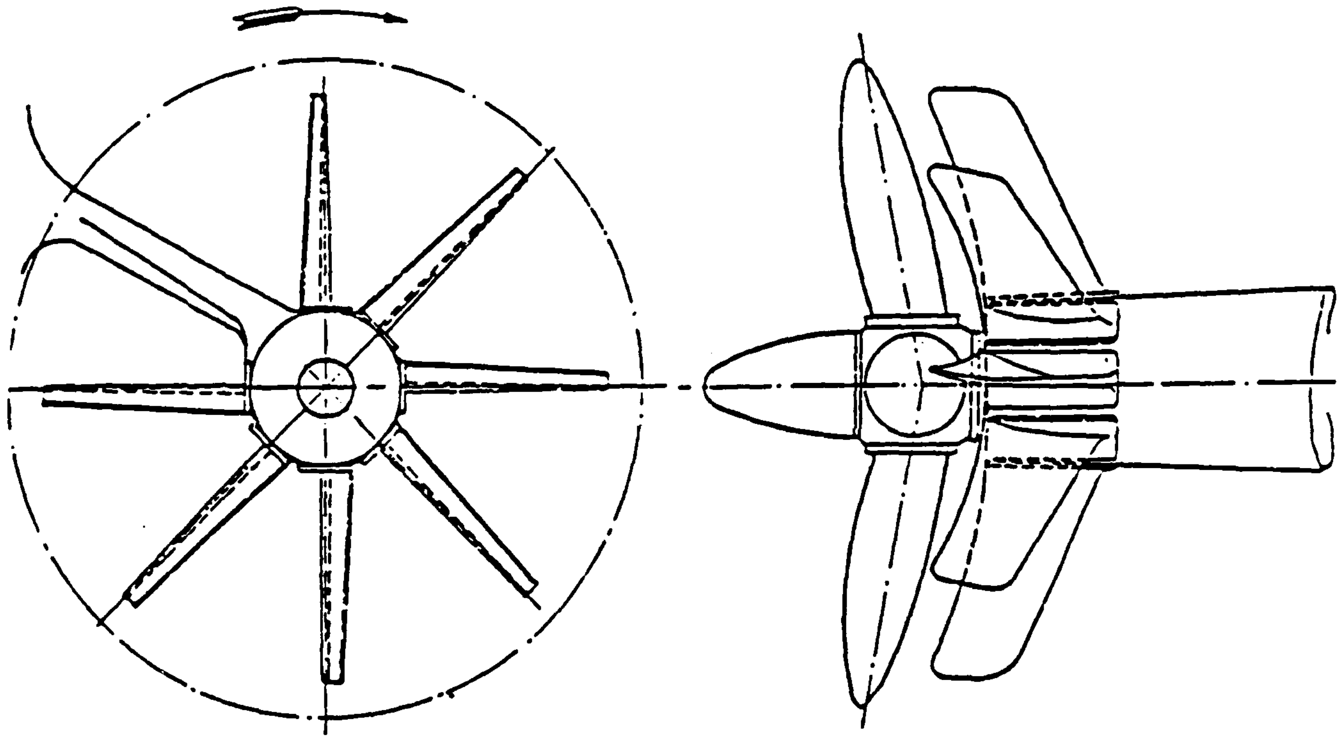


FIGURE 1-1: SYMMETRIC PRE-SWIRL VANES
(GERMAN PATENT, 1912)

2.0 THEORY OF PRE-SWIRL VANES

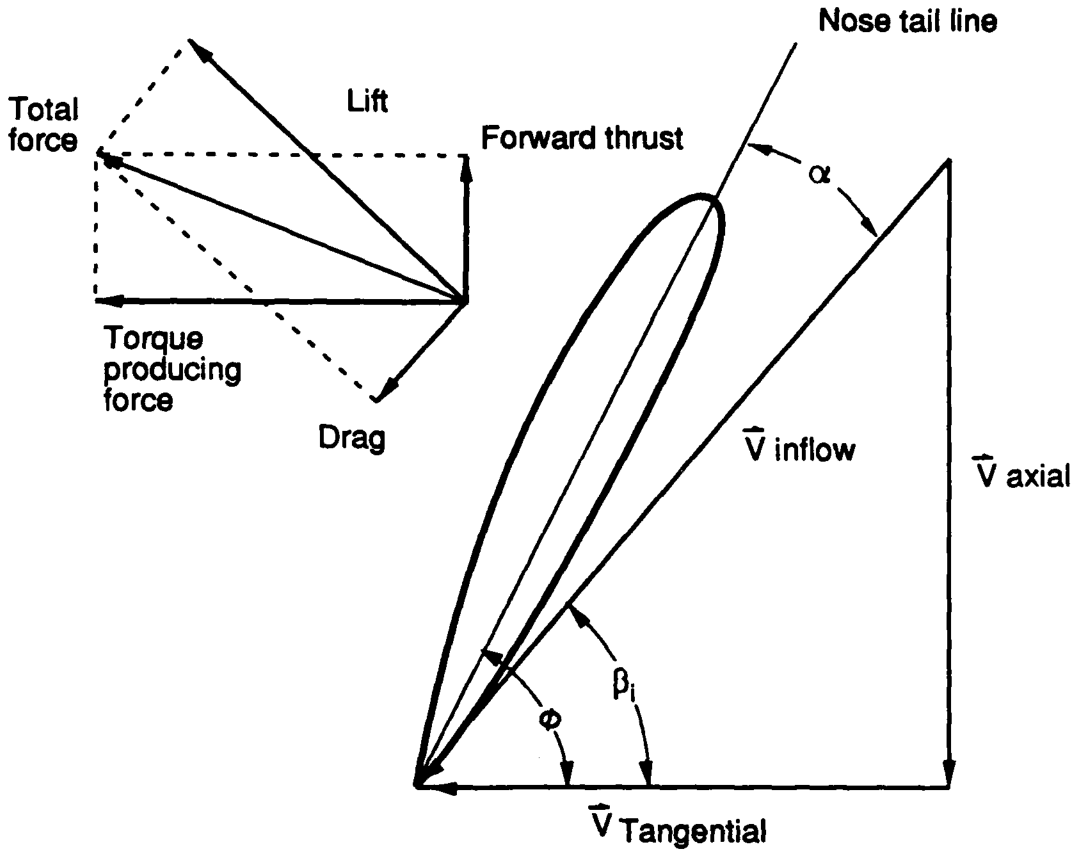
Rotational energy in the slipstream behind a propeller provides no useful thrust and therefore detracts from propeller efficiency. These losses increase with increasing propeller load coefficient, $C_T = T/(\rho A V_a^2)$, so pre-swirl vanes will provide greatest efficiency improvements when C_T is large, as in icebreakers, tugboats, mine sweepers, etc. Numerous devices have been proposed to reduce this energy loss, such as asymmetric sterns, contra-rotating propellers, contra-guide rudders, and pre- or post-swirl vanes.

Before discussing vane design, it is instructive to review the components of the inflow velocity seen by a conventional propeller. As shown in Figure 2-1, there is an axial velocity from the vessel's forward speed and a tangential velocity from the propeller's rotation. These axial and tangential velocities are modified in two ways. First, the presence of the hull ahead of the propeller creates a wake effect by modifying the potential flow, and developing a viscous boundary layer.[1] Secondly, the action of the propeller itself induces both axial and tangential velocities in the flow.[2] Thus, there is a net hydrodynamic inflow velocity at angle β_i. The difference between β_i and the

---

[1]These hull-induced effects are traditionally expressed as axial and tangential wake fractions, W_x and W_t Thus, the axial velocity behind a towed hull is V_s^x $(1-w_x)$ and the tangential velocity is $2\pi nr(1-w_t)$. The quantities w_x and w_t vary throughout the plane of the propeller, and are thus functions of the propeller radius, r, and blade angle, Θp. They are usually determined by wake surveys behind a towed model, and are sometimes corrected for scale effects.

[2]These axial and tangential propeller-induced velocities are traditionally expressed by self-induction factors, a and a', respectively. Thus the axial and tangential velocities behind a self-propelled vessel are the same as those behind a towed vessel, but modified by the factors (1+a) and (1-a'), respectively. The quantities a and a' can also vary according to their location in the propeller plane, and are thus functions of radius and blade angle. They are usually calculated by lifting line or lifting surface theories.



$V_{axial} = V_S\ (1\text{-}w_X)\ (1 + a)$ where:

V_S = ship speed
$(1\text{-}w_X)$ = axial wake effect factor
$(1 + a)$ = self-induced axial velocity factor

$V_{tangential} = 2\pi\, nr\ (1\text{-}w_t)\ (1 - \acute{a}) + v_2 \sin \theta p$ where:

$2\pi\, nr$ = velocity due to propellers' own rotation
$(1 - w_t)$ = tangential wake effect factor
$(1 - \acute{a})$ = self-induced tangential velocity factor
$v_2 \sin \theta p$ = inclined shaft effect (see Fig. 2-2a)

(Note: W_X and W_t vary with r and θp)

FIGURE 2-1 Inflow Velocity Diagram for Conventional Propeller

blade pitch angle is the effective angle of attack. Since the wake fractions and self-induced velocities vary with θ_p, both the inflow angle and therefore the angle of attack changes as the blade section rotates.

If the propeller operates on an inclined shaft, there will also be an apparent upflow in the plane of the propeller, as shown in Figure 2-2. Thus, as the blade rotates, it will experience a sinusoidal variation in tangential velocity due to this apparent upflow. This further contributes to variations in the angle of attack.

In conventional propeller design, it is generally desirable to develop the most efficient radial distribution of circulation, [3, 4]. It is also desirable to have the blade sections operate (as nearly as possible) at their "ideal angle of attack"[3].

Commonly used propeller blade sections are shaped so that, at this ideal angle of attack, the pressure distribution on the suction side is nearly uniform along the chord. When the angle of attack varies, as discussed above, the low pressure will peak near the leading edge, as suggested by Figure 2-3.

If this pressure is below the vapor pressure of water, a vapor cavity is created. When these cavities are convected downstream into a higher pressure region, they implode violently causing noise, vibration and sometimes erosion of the blade or rudder. This tendency towards cavitation can be avoided by increasing depth of propeller which is often impractical, or by increasing the chord lengths. Increased chord lengths, however, increase the frictional drag on the blade.

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[3] The "ideal angle of attack" is that angle at which the velocity at the leading edge remains finite according to potential flow theory. This is sometimes also referred to as "shock-free entry". Since the section angle of attack is thus fixed, the only way to develop the required lift is to use a "cambered" section (asymmetrical about nose-tail line) as shown in Figure 2-2.

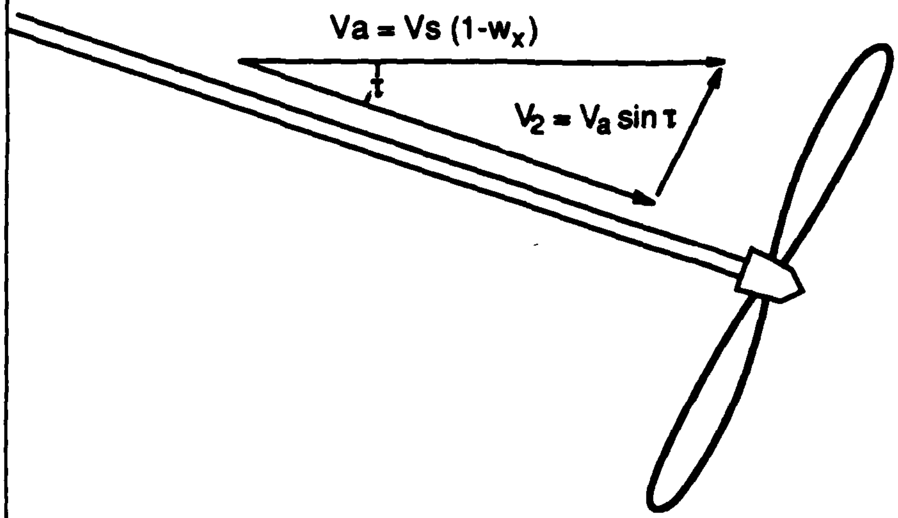
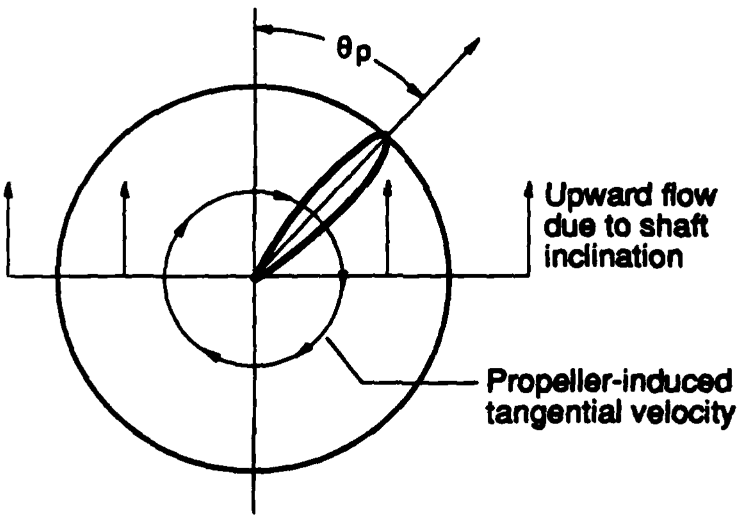


FIGURE 2-2a Apparent Vertical Upflow in Plane of Propeller due to Inclined shaft

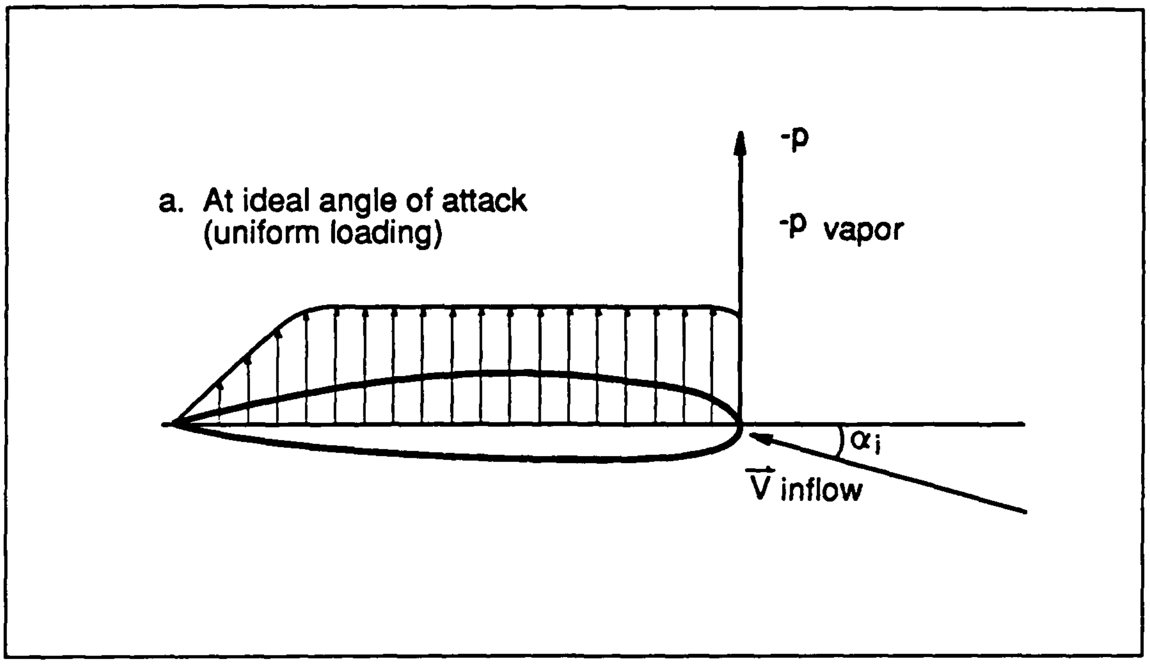


Tangential velocity = $2\pi nr\,(1 - a')\,(1 - w_t) + v_2 \sin \theta p$ where:

$2\pi nr$ = velocity due to propellers' own rotation
$(1 - a')$ = self-induced tangential velocity factor
$(1 - w_t)$ = tangential wake fraction factor
$v_2 \sin \theta p$ = component of v_2 in tangential direction

FIGURE 2-2b Effect of Vertical Upflow on Velocity Tangent to Propeller Rotation

FIGURE 2-2 Effect of Inclined Shafts on Tangential Velocity



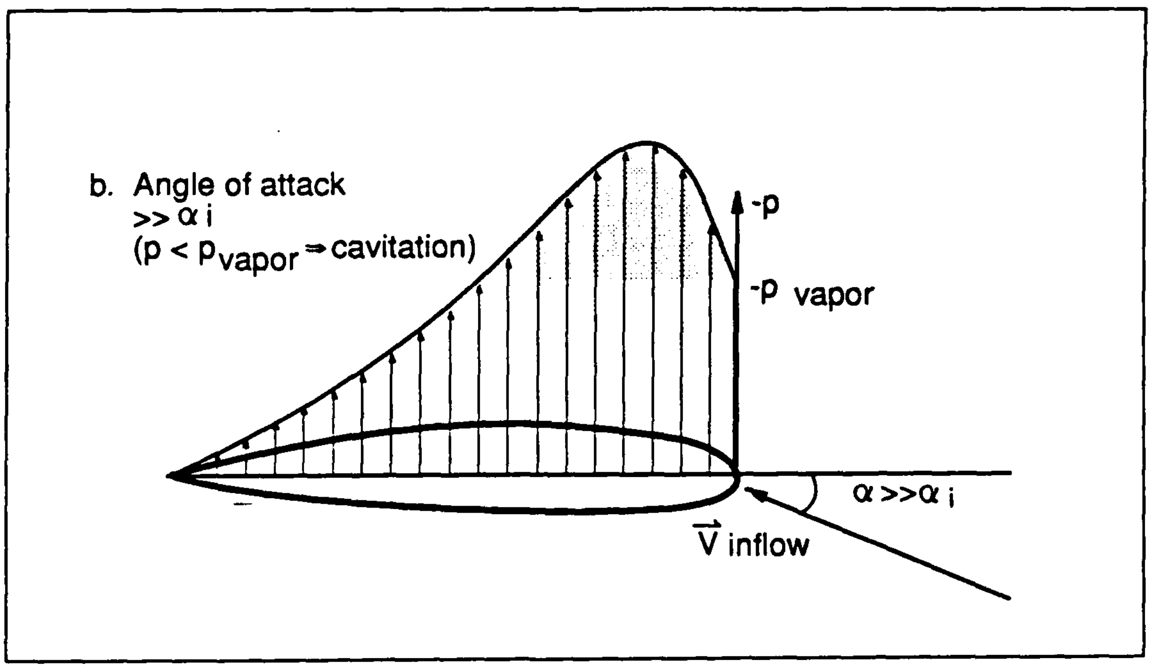


FIGURE 2-3 Effect of Angle of Attack on Chordwise Pressure Distribution

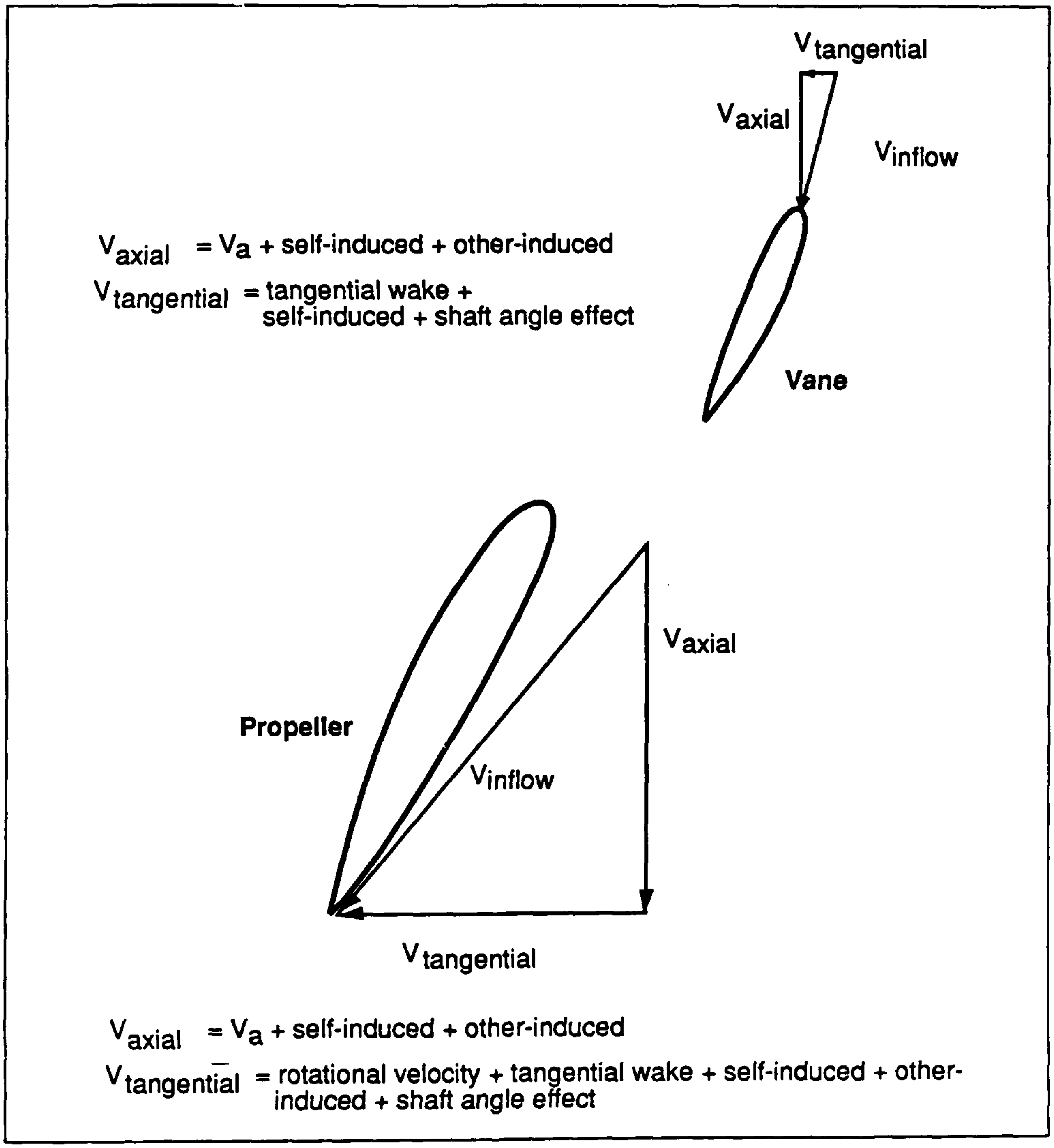


FIGURE 2-4 Velocity Diagram for Combined Vane / Propeller System

A propeller operating behind a pre-swirl vane set will experience additional axial and tangential velocities induced by the vanes. The vanes will experience an axial inflow, axial and tangential wake effects, a self-induced axial and tangential velocity, and a propeller-induced axial velocity. (The propeller induces no tangential velocity on the vanes, according to Kelvin's theorem). This is illustrated in Figure 2-4.

In uniform horizontal flow, symmetric vanes should be selected to cancel the rotational losses in the wake far downstream. Thus they should induce a rotational velocity on the propeller equal and opposite to the propeller's self-induced tangential velocity.

When wake variations or upward flow due to an inclined shaft cause varying inflow angles to the propeller sections, asymmetric vanes can be designed to reduce this variation and prevent cavitation. Figure 2-2a shows the upward flow due to an inclined shaft. On the left half of the disk, Figure 2-2b the upward flow is in the same direction as the propeller-induced tangential velocity. Since both velocities are in the direction of rotation, they reduce the tangential velocity seen by the blade (see Figure 2-1). This increases $_i$, reduces a , and can lead to face cavitation.

On the right half of the disk in Figure 2-2b, however, the upward flow opposes the propeller-induced velocity. This increases the tangential velocity seen by the blade in Figure 2-1, increases a, and may lead to back cavitation as shown in Figure 2-3. Returning to Figure 2-2b, it is seen that counter-clockwise pre-swirl on the left half of the disk would (a) prevent the decrease in angle of attack, and (b) cancel the propeller-induced tangential velocity which causes rotational energy losses. On the right half of the disk, the direction of

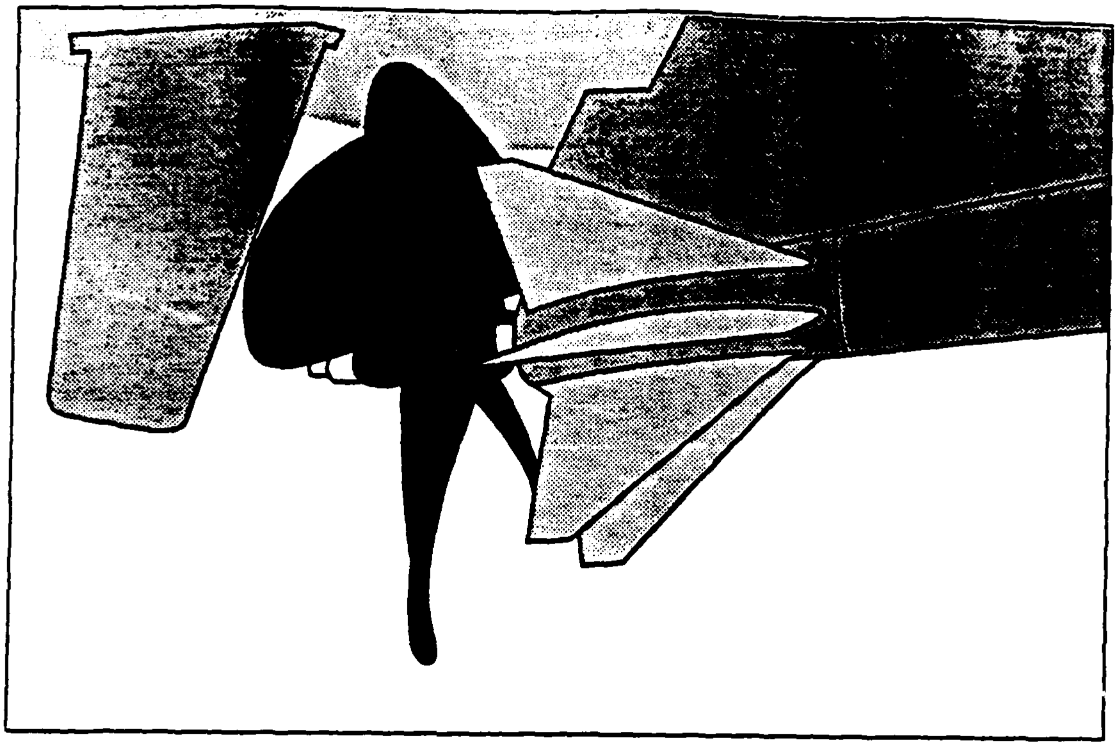


FIGURE 2-5a: VIEW OF VANES AND PORT PROPELLER (LOOKING OUTBOARD)

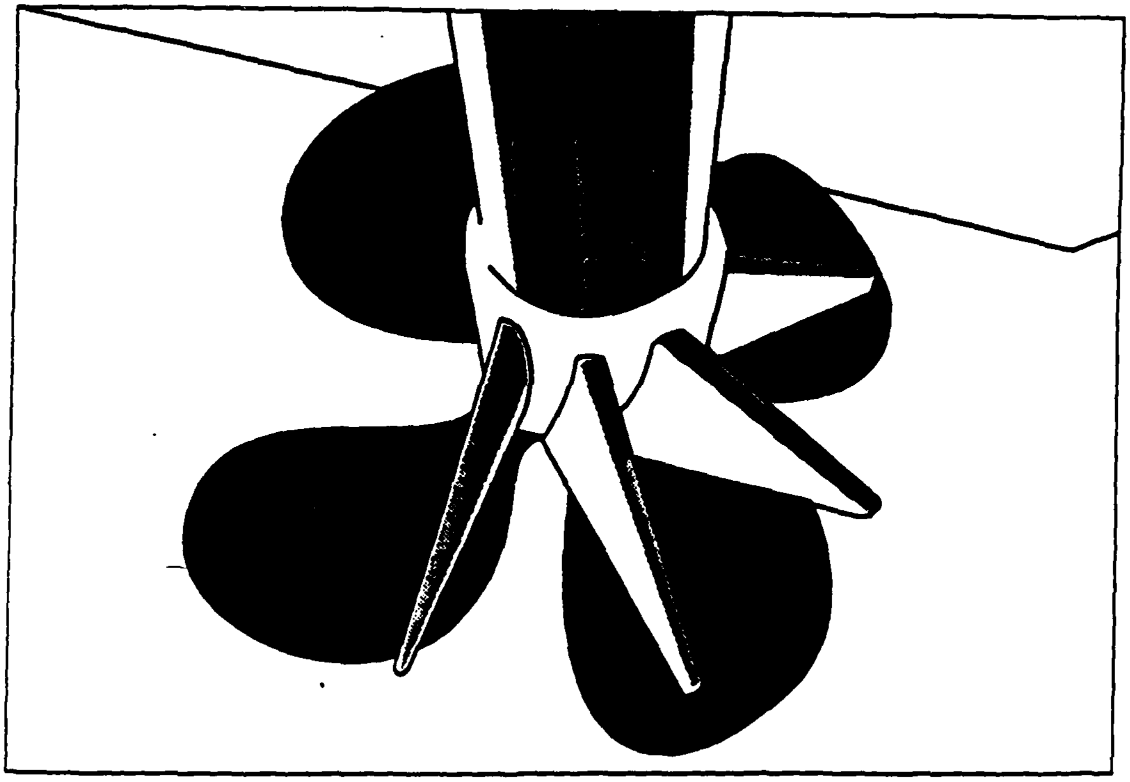


FIGURE 2-5b: VIEW OF VANES AND STARBOARD PROPELLER (LOOKING AFT)

FIGURE 2-5: PENN STATE'S ASYMMETRIC VANE DESIGN

(See Appendix B for additional propeller data)

desired pre-swirl velocity depends on the relative magnitudes of the upflow and propeller-induced velocity. If these velocities are nearly equal, it may be advantageous to omit these vanes entirely, leading to the asymmetric configuration shown in Figures 2-5 and B-1.

Thus, the pitch and camber distributions of the vane set must be carefully matched to those of the blade section downstream to provide the ideal inflow angle to the blade section, while cancelling just the right amount of rotational energy. (The radial distributions of lift on the vanes and propeller also influence efficiency, so the pitch and camber distributions chosen should also develop these optimal lift distributions). The vanes will, of course, create additional drag, and each application must be evaluated to determine if the efficiency improvement from the reduction of rotational energy loss outweighs the additional drag.

It is, of course, difficult to specify the vane geometry which will achieve these diverse goals simultaneously. Too much or too little pre-swirl will change the angle of attack, and create the possibility of unacceptable amounts of cavitation. Thus, accurate methods of predicting the flow around vanes and propellers are absolutely necessary in order to realize the advantages that vanes can offer.

In summary, an optimal vane/propeller combination should:

a) provide optimal radial load distributions on both the propeller and vanes.

b) provide (as nearly as possible) a constant (ideal) angle of attack to the propeller blade sections so that a more uniform chordwise load distribution can be maintained as the blade rotates.

c) cancel the rotational energy so that the flow far downstream from the propeller is purely axial.

d) create minimum drag and torque-producing forces, and predict their magnitudes, so that rated engine HP and RPM will be achieved.

Penn State uses a "streamline curvature" method [5], to design the propeller and vane geometry. Their first designs for the Coast Guard 41' utility boat (UTB), however, developed excessive RPM, and severe cavitation was reported in field trials [1]. Larimer, et al., then experimented with other propellers (greater pitch, larger diameter, different radial load distributions). By trial and error, a propeller was found which, even without the vanes, reportedly improved the performance of the baseline propeller by up to 10%, depending on the boat's speed. With the vanes, further improvements of up to 5% were found. It is unlikely, however, any of these configurations were truly optimum in the sense of the design criteria presented earlier. It is clearly necessary to develop a more rigorous design method, so that such costly experiments will not be needed for future designs.

References [6] and [7] describe lifting line theories for design of optimal multi-component propulsors. The propeller is modeled by classical lifting line theory with helical shaped trailing vortices. The vane set is also represented by lifting line theory, but with its trailing vortices streaming straight aft. Circumferential averages of the axial wake at each radius can be input. Both methods then use an optimization technique to find the radial distributions of circulation which produce the required thrust at a specified RPM with minimal torque. The self-induced and other-induced velocities on the vane and propellers can then be calculated from lifting line theory. These velocities are then used to estimate the pitch and camber distributions required for both vanes and propeller blades to operate as nearly as possible at their ideal angles of attack. Lifting surface methods should then be used to determine the final geometry, since three-dimensional effects are significant for low aspect ratio blades. Reference [8] reports on the design of a symmetric vane set using this procedure, and water tunnel tests showed gains in efficiency up to 8% (after accounting for stator drag), even higher than predicted.

For an inclined shaft, Kerwin et. al. [7] assume that the circulation distributions found for the symmetric inflow case will still be optimal in a circumferentially averaged sense. They then proceed to add harmonics to the circulations (which make no contribution to the average circulation) but can be used to construct a Fourier series representation of the circulation which will cancel the variations caused by the apparent upflow due to the inclined shaft. This design method is very promising, but depends strongly on its assumptions of trailing vortex geometry.

The Coast Guard has recently commissioned MIT to do such theoretical design calculations for several propeller/vane combinations, including the Penn State design shown earlier in Figure 2-5, and one based on their own optimization method. It is hoped that these new geometries can also be tested to provide data on the validity and limitations of the current theory. Though the design methods for propeller/vane combinations are clearly more complicated, there appears to be no reason why the theories which have been so successful in conventional propeller design cannot be extended to propeller/vane combinations.

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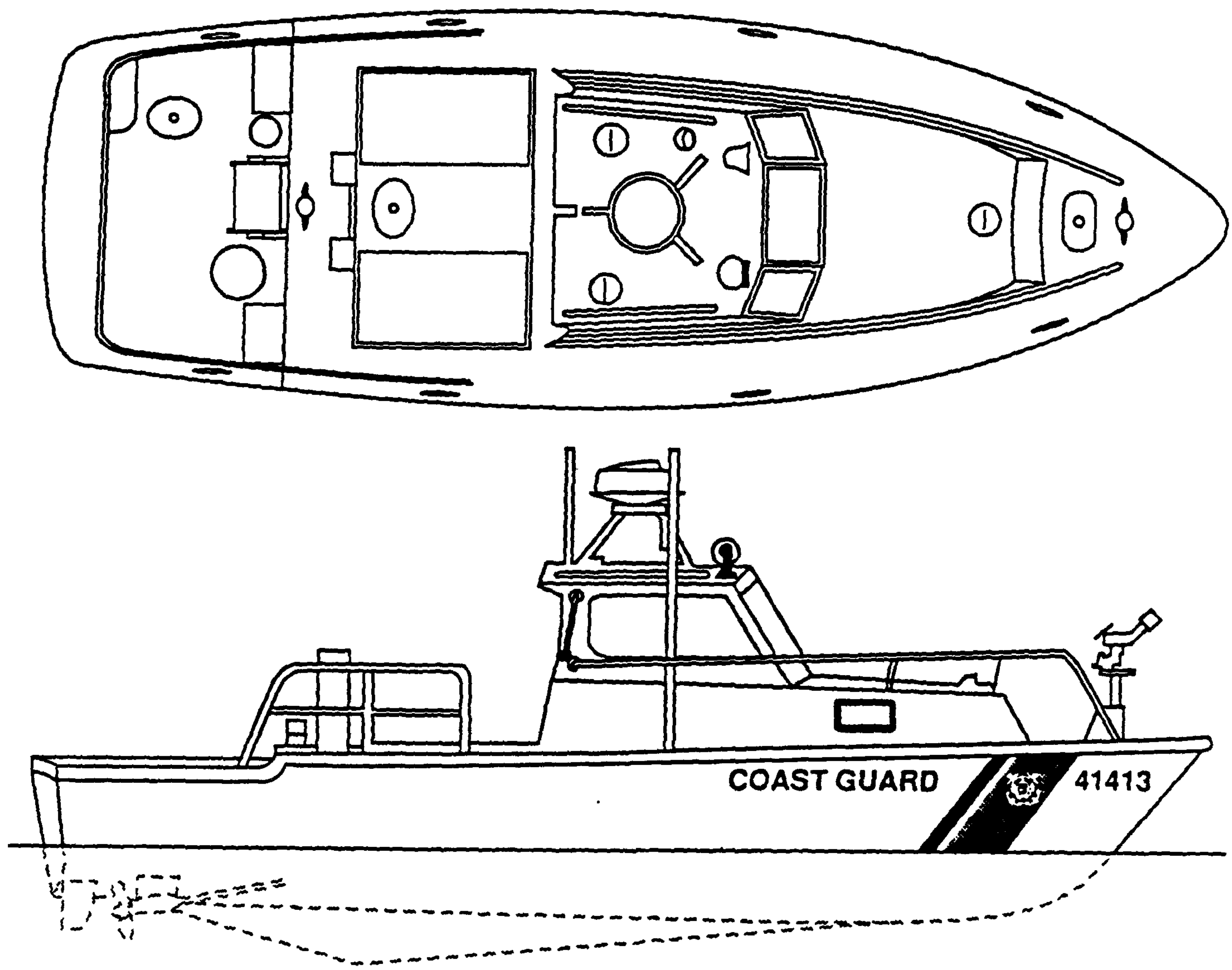
3.0 PREVIOUS TEST RESULTS

A research project was begun in 1985 to determine if pre-swirl vanes could further improve the fuel efficiency of the Coast Guard fleet. Penn State's Applied Research Laboratory was commissioned to design an asymmetric vane set and matching propeller (Figure 2-5) for the Coast Guard's 41-foot twin screw utility boat (UTB) (Figure 3-1). Reference [1] describes the initial tests of this design. The new vane/propeller combination was found to be less fuel efficient than the original constant-pitch baseline (BL) propeller. Also, at full throttle, the rated RPM of the engine was exceeded, and audible cavitation noises were reported. Tests were then conducted with the original BL propeller and the new vane set, and showed improved fuel consumption relative to the BL propeller alone. Additional tests [9], were conducted in 1986 to study the effects of propeller/vane interaction, and attempt to further optimize the configuration. A set of new modified stock propellers (MSP), with a linearly increasing pitch distribution from root to tip, was fashioned by re-pitching a pair of commercially available stock propellers. These propellers, without the vanes, performed substantially better than the original baseline propellers alone. When the vanes were added, the MSP with vanes proved to be the most efficient of all the configurations tested. (Details of all the propellers tested are given in Appendix B). The results of these tests are summarized by Figure 3-2 from reference [1].

Despite these very promising results, questions remained about the accuracy of the turbine-type fuel meters used, and the possible effects of different trim angles and engine operating points with the various propulsors. It was decided to re-test the baseline and modified stock propellers, with and without vanes, on a different 41' UTB. A positive displacement-type fuel meter was used, and torque was measured by standard strain guage rosettes bonded to the shafts. The remainder of this paper



Length = 41'

Beam = 13.5'

Draft = 4'

Design Displ. = 28,600 lb
(full load - no passengers)

Test Boat Actual Weight = 30,700 lb

Twin Screw

Twin Cummins VT 903 m diesels
(rated at 318 SHP, 2600 Engine RPM)

FIGURE 3-1: 41' UTILITY BOAT, PROFILE AND PLAN VIEWS

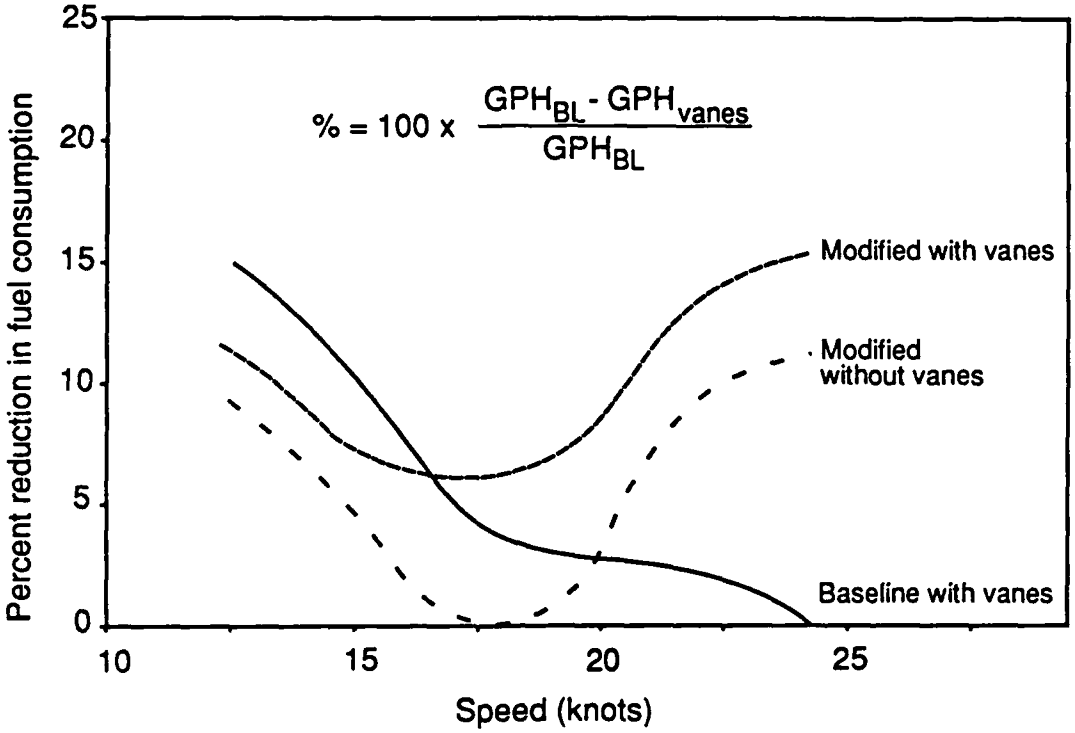


FIGURE 3-2 Fuel Savings Relative to Baseline Propellers from 1986 Tests (reprinted from [1])

will describe the methodology and results of this test series. A cost/benefit analysis is then presented to determine whether the measured savings warrant fleet-wide installation of the new vane sets on all 41' UTBs.

4.0 TEST DESIGN AND EXECUTION

The tests included measurements of vessel speed, fuel consumption, shaft horsepower, RPM, and the vertical vibration of the hull plating above the propeller. These tests were conducted on a different 41 ft. UTB of the same design and similar weight. The baseline propellers, therefore, may have been different than those used in the earlier tests [1], but the modified stock propellers were identical. Both the baseline and the modified stock propellers were tested with and without vanes. In addition to the main test program, tests were also conducted at a lighter displacement and the fuel consumption of a second 41′ UTB (boat #41309) was measured at various trim angles induced by stern wedges (see Appendix D). All tests conducted are summarized in Table 4-1.

After the completion of these instrumented trials, the MSP and vanes will remain on boat 41413 for a one-year operational evaluation to determine susceptibility to fouling, damage from debris or grounding, and cavitation.

4.1 Water Depth and Test Site

References [10] and [11] were consulted in an effort to determine what shallow water effects might be present at a 30 foot mean low water depth with four foot tidal variations. Reservations concerning the applicability of these studies led to the decision not to correct the test data for shallow water effects. Whatever added resistance effects there were, are therefore present in the data. The test course is illustrated in Figure 4-1, showing the ample space available to get the boat up to speed and to stabilize the fuel rate. Also, good protection from light chop was afforded by the breakwater, allowing operation on either side, depending on the wind direction. Testing was suspended when the chop exceeded 6 inches, usually associated with wind in excess of 15 knots.

TABLE 4-1

COMBINATIONS OF TESTS PERFORMED

	UTB 41413
Baseline Props (BL)	Tests 101, 102, 103, 104, 109
Modified Stock Props (MSP)	Tests 105, 106, 107, 108
Modified Stock Props & Vanes (MSPV)	Tests 110, 111
Baseline Props & Vanes (BLV)	Test 112
	UTB 41309
Baseline Props (BL)	Tests 201, 202
Modified Stock Props & Wedges (MSPW)	Test 203

4.2 Fuel Measurement

A single, positive displacement fuel flow meter was chosen to record the amount of make-up fuel added to the recirculating fuel system. Care was taken to cool and settle the fuel before measuring the flow volume. The meter was calibrated at the factory, checked in the R&DC laboratory and checked again by use of a day tank on the boat. This meter proved to be both accurate and consistent. Details of this system are discussed in Appendix A.

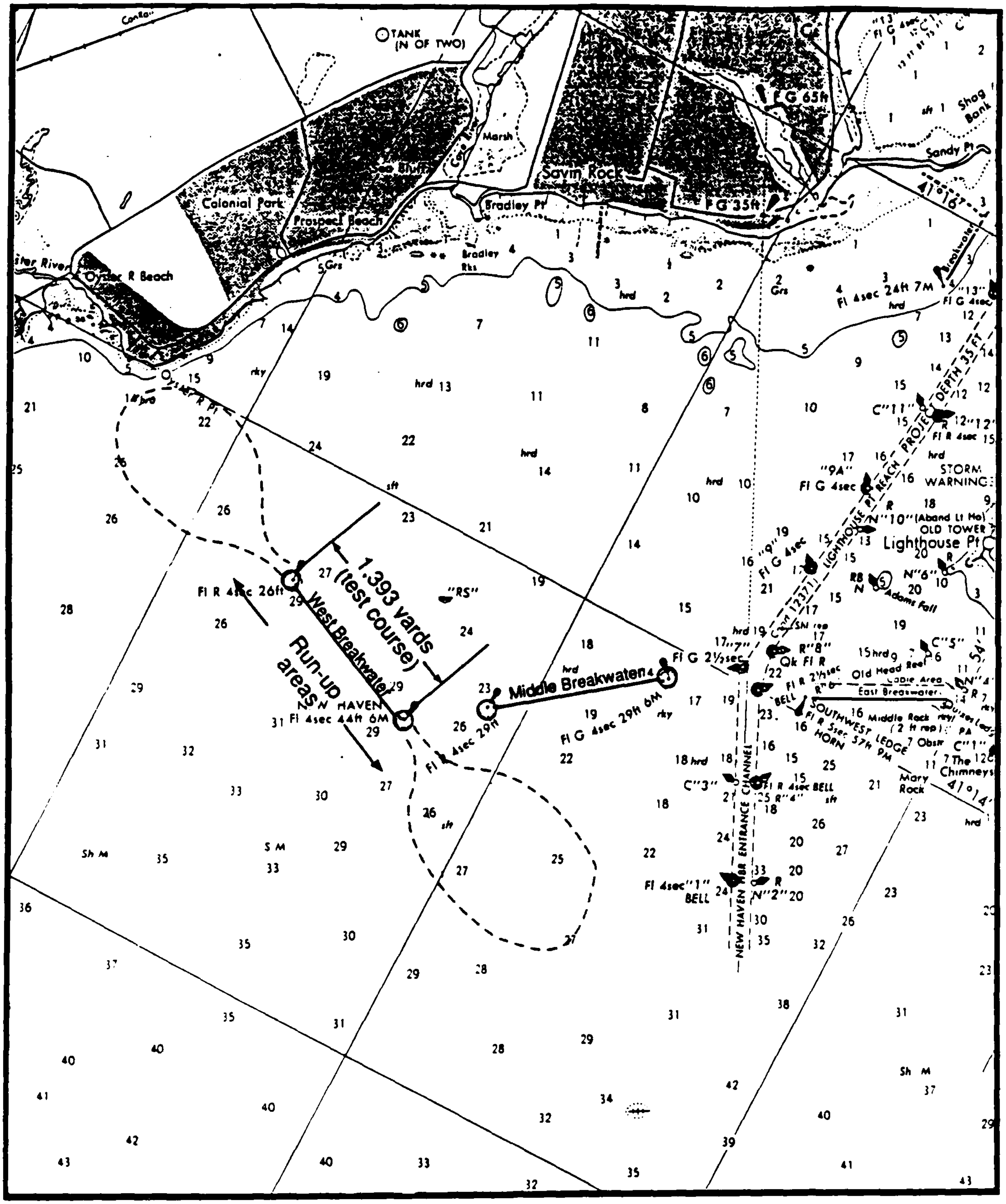


FIGURE 4-1 Test Site — 41 ft UTB, New Props & Vanes, West Breakwater, New Haven, CT

4.3 Speed Measurement

Measured time over a fixed distance was chosen as the most reliable measurement of speed. The distance was determined from surveyed positions and the time was measured by stopwatch. Care was taken to place the observer with the stopwatch in exactly the same position on the boat and to keep the boat on a constant heading, especially at the beginning and end of the test run. Each run consisted of two legs, in opposite directions, so that current effects could be averaged out.

4.4 Torque, RPM, Vibration and Trim Measurement

Torque was measured by calibrated strain gauges applied to the shafts. The calibration was performed by applying weights to a moment arm as described in Appendix A. The RPM and torque signals were transmitted from the shafts and processed in the forward cabin. Readings were taken visually, with tape-recorded backup. The RPM signals were calibrated using a hand-held tachometer. Accelerometers were placed on the plating above the propellers to detect any significant change in vibration from the new props or the prop-vane combination. Trim was measured by bubble inclinometer and occasionally was checked by an electronic inclinometer signal to the tape recorder.

4.5 Data Processing

Measurements were made at seven different RPM targets to obtain enough data to plot faired curves. Each data point is the average of two legs, down and back on the course. The speed, fuel consumption, RPM, torque and horsepower were determined separately for each leg, and then averaged over the two legs. These values are plotted, and faired curves plotted through the points. When determining the percent savings in fuel and horsepower, the data were least-squares fit to a 4th order polynomial. Correlation coefficients were all .999 or higher,

indicating an excellent fit to the data. These polynomials were then used to interpolate results with different propulsors to a common speed where percentage differences could be computed. This curve-fitting eliminated any possible ambiguity in hand-fairing the plots and reading faired values from a graph.

4.6 Additional Test Considerations

During the initial tests, the port and starboard RPMs were set by adjusting the throttles until the boat's tachometers showed the desired readings. However, when the port and starboard RPMs were matched on the boat's tachs, the RPMs recorded by our measurement system varied significantly. A hand-held strobe was then used to check the shaft RPMs, and gave results nearly identical to the measurement system. Thus, although the boat's tachs showed identical RPMs, in fact the starboard engine was turning 120 RPM faster than the port engine at top speed. The port engine fuel stop was then advanced to enable it to develop 2600 RPM. In all subsequent tests, the desired RPMs were matched on the measurement system rather than the boat's tachometers. Results from these early tests are not reported here because of the mis-matched RPMs. Nonetheless, fuel and HP measurements from these early tests showed little difference from those obtained later with matched RPMs. These early data are available at the R&D Center if desired.

When the BL propellers were removed, it was noted that they were apparently from different manufacturers, had visibly different hub shapes, and differed in weight by 34%. Since no matched set was immediately available and the schedule prohibited re-testing, these BL propellers were retained for further tests with the vanes.

It is standard Coast Guard procedure to inspect the bilges periodically for flooding. During the night preceding the first test of the baseline propellers with matched RPMs, this

inspection was performed during a heavy rainstorm. One strain gauge became wet, resulting in erratic port torque readings during the testing of the BL props. Since the schedule made it impossible to repeat these tests, port HP's were estimated in the following manner. Calculations were performed using the results of Troost's B-series model tests, with an assumed wake fraction of 0.08. These calculations were made at the RPMs measured on the port shaft during the tests with the mismatched RPMs. A correction factor, the ratio of calculated to measured HPs, was then plotted as a function of the advance ratio, J. Similar calculations were then made at the port RPMs measured during the matched RPM tests with the BL props. The correction factors determined from the mismatched RPM tests were then applied to the calculated HP's. One should naturally regard these results with some skepticism, but the agreement between the HP and fuel consumption data, to be discussed later, suggests that these estimated port HP data are reasonable.

During the pre-vane tests, it was noted that the fuel metering system provided marginal cooling capacity, and slow speed cooling-off periods were required between test runs. It was decided to increase the pump capacity when the vessel was hauled for installation of the vanes. This was done to increase the flow of cooling fluid and did not affect the supply to the engines, which is taken from a vented tank (see Figure A-2). This change does not affect the measured fuel consumption, and is not considered to introduce any significant systematic error.

The vessel was then hauled and cradled, so the vanes could be fitted. This required removing the strut bearings so they would not be destroyed by the heat from welding. This required removing the shafts, and therefore the original strain gauge system. New gauges were installed on both shafts after the vanes had been welded in place, and both shafts were re-calibrated as described in Appendix A. During the calibration, it was noted that the starboard meter did not quite return to zero after

unloading. This non-zero reading was recorded at the start and end of each day's tests, and the average was subtracted as a tare from the port torque readings before applying the calibration factors. The possibility of introducing a systematic error by using different gauges and different calibrations for the tests with vanes cannot be entirely dismissed. Again, however, the agreement between HP and fuel consumption data, (to be discussed) suggests that any such errors must have been small.

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5.0 TEST RESULTS

The results of the tests on vessel 41413 are summarized in Figures 5-1 through 5-7. Numerical values for the plotted points are given in tables in Appendix C.

It is evident that the results from the earlier tests shown in Figure 3-2 are not consistent with those from these tests shown in Figure 5-7. Questions naturally arise about the accuracy and repeatability of both series of tests. Nonetheless, the independent measurement of fuel consumption and horsepower provides some means of checking the consistency of the results. Figure 5-8 shows an engine fuel map, obtained from manufacturer's tests with No. 2 diesel oil at ISO standard temperature and pressure. It can be observed from the data points plotted on this figure that the differences in specific fuel rates between the various propulsors at equal RPM are generally on the order of 1%. Thus, a certain percentage reduction in HP should be reflected by a nearly equal percentage reduction in fuel rate. Since these measurements are completely independent of each other, the congruence of HP and fuel rate data provides a valuable check on the quality of the data. While Figures 5-6 and 5-7 are not identical, they show the same relative rankings for the various propulsors, the same convex shape, and similar maximum savings. In view of the various difficulties with the torque meters described earlier, it is suggested that the figures for percent reduction in fuel rate are probably more reliable than those for HP. In all cases, we are looking at relatively small differences, and the reader is advised to attach more significance to the trends and qualitative comparisons than to the precise magnitudes of the reductions.

It should be stated that the tests in reference [1] showed a similar self-consistency between their percent reductions in HP and fuel rate. One possible explanation for the differences between Figure 3-2 and Figure 5-7 is that the baseline propellers used in the earlier test may have been substantially different

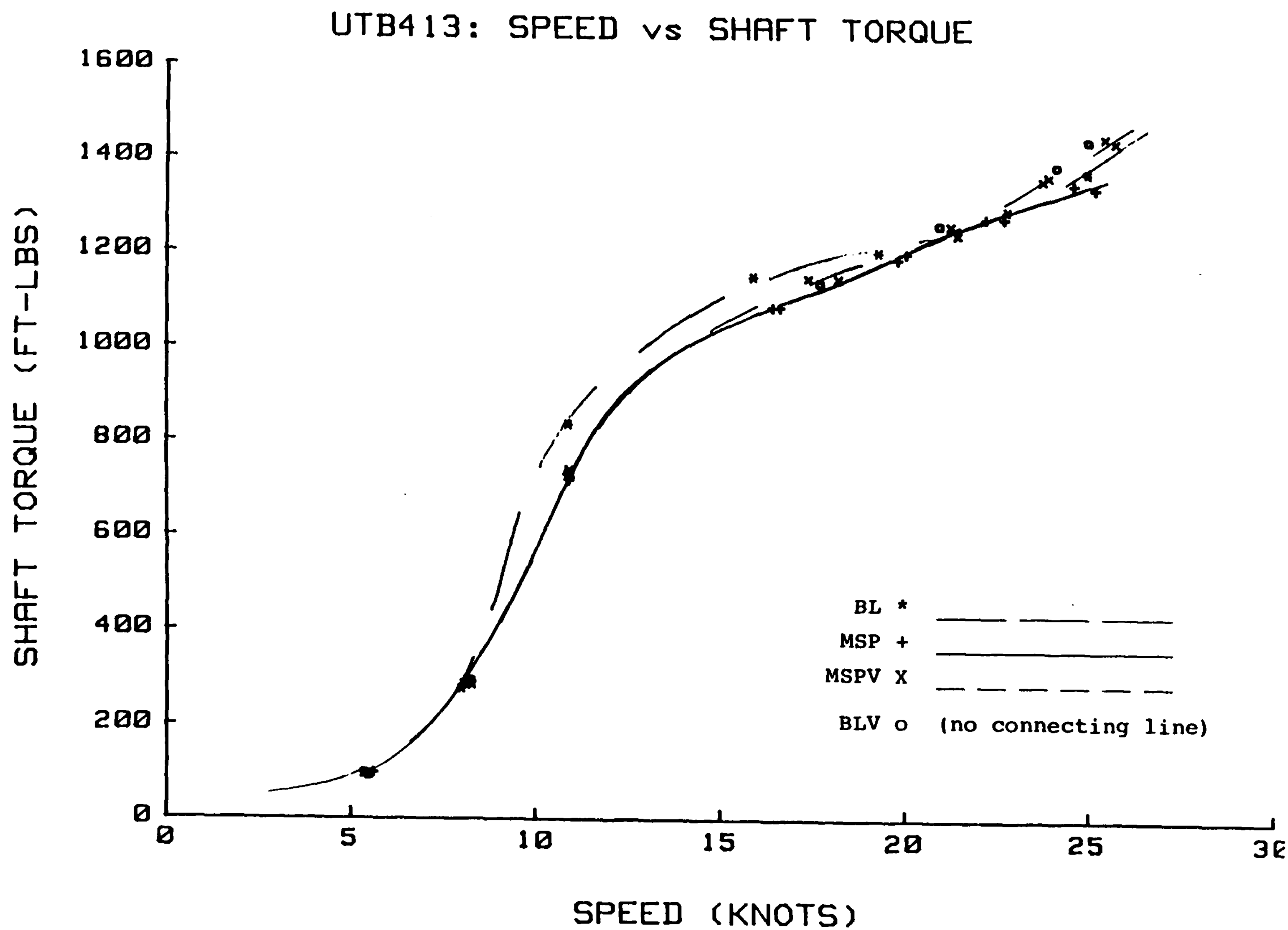

FIGURE 5-1: UTB 413 - SPEED vs SHAFT TORQUE

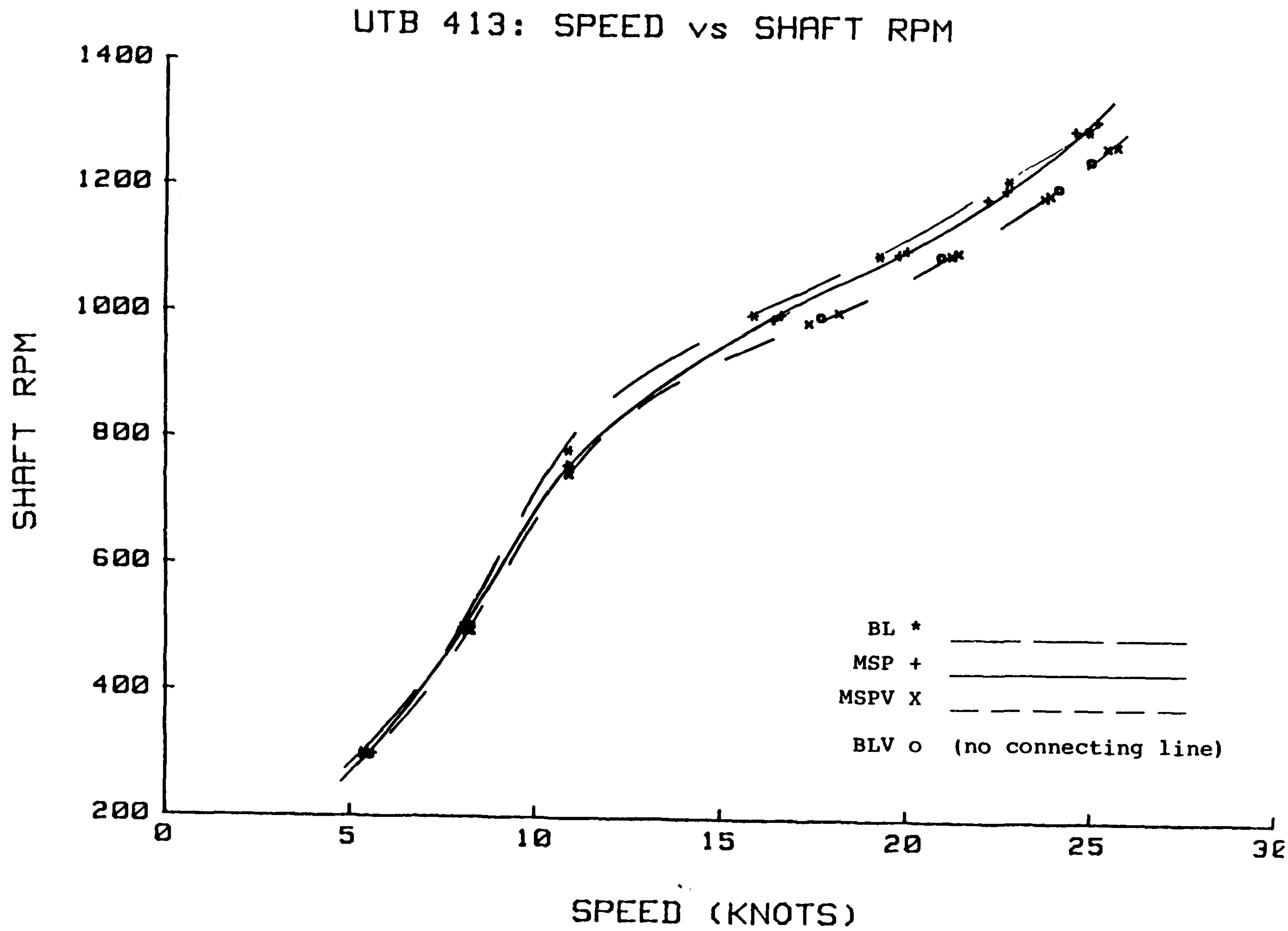

FIGURE 5-2: UTB 413 - SPEED vs SHAFT RPM

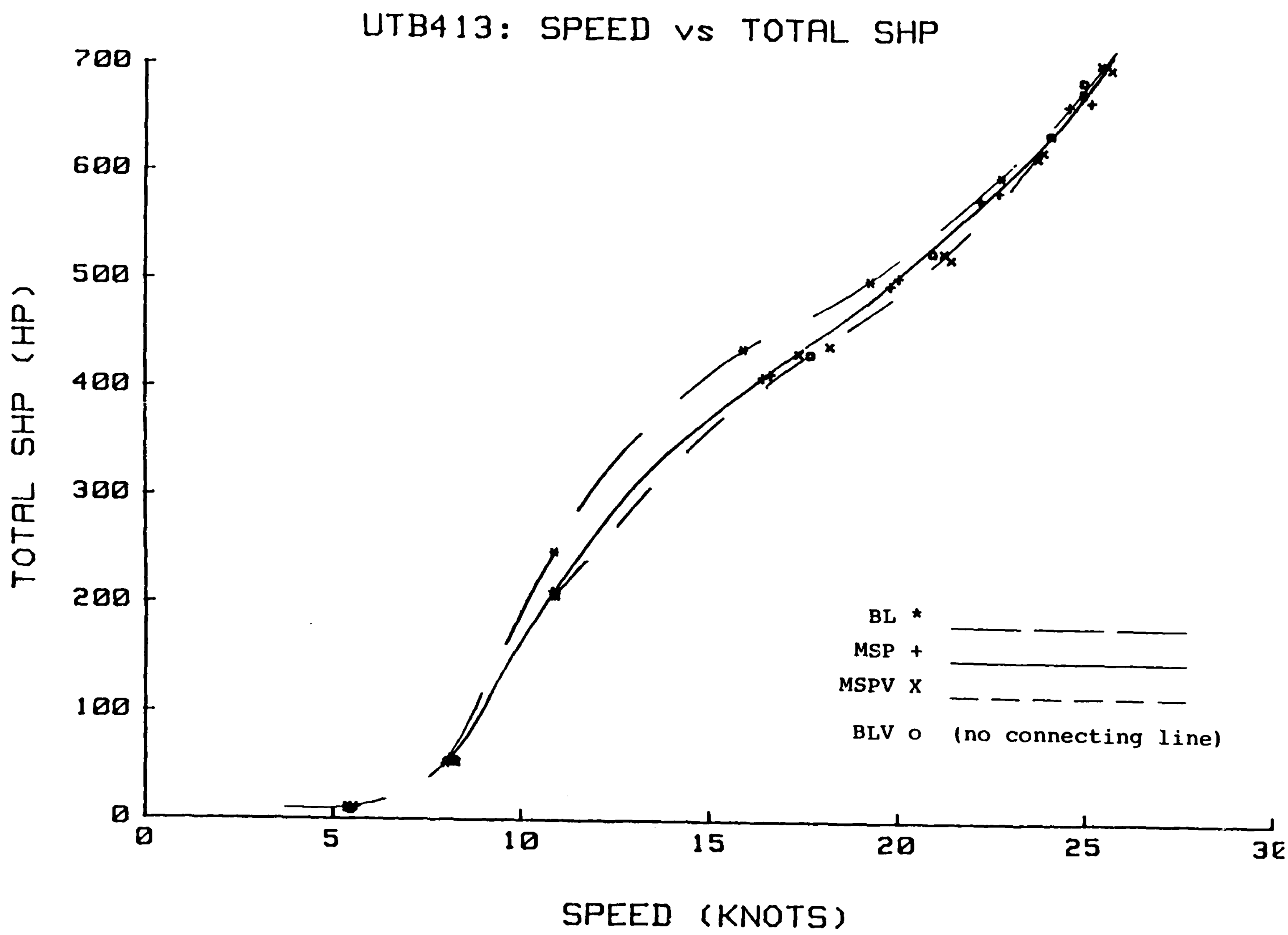

FIGURE 5-3: UTB 413 - SPEED vs TOTAL SHP

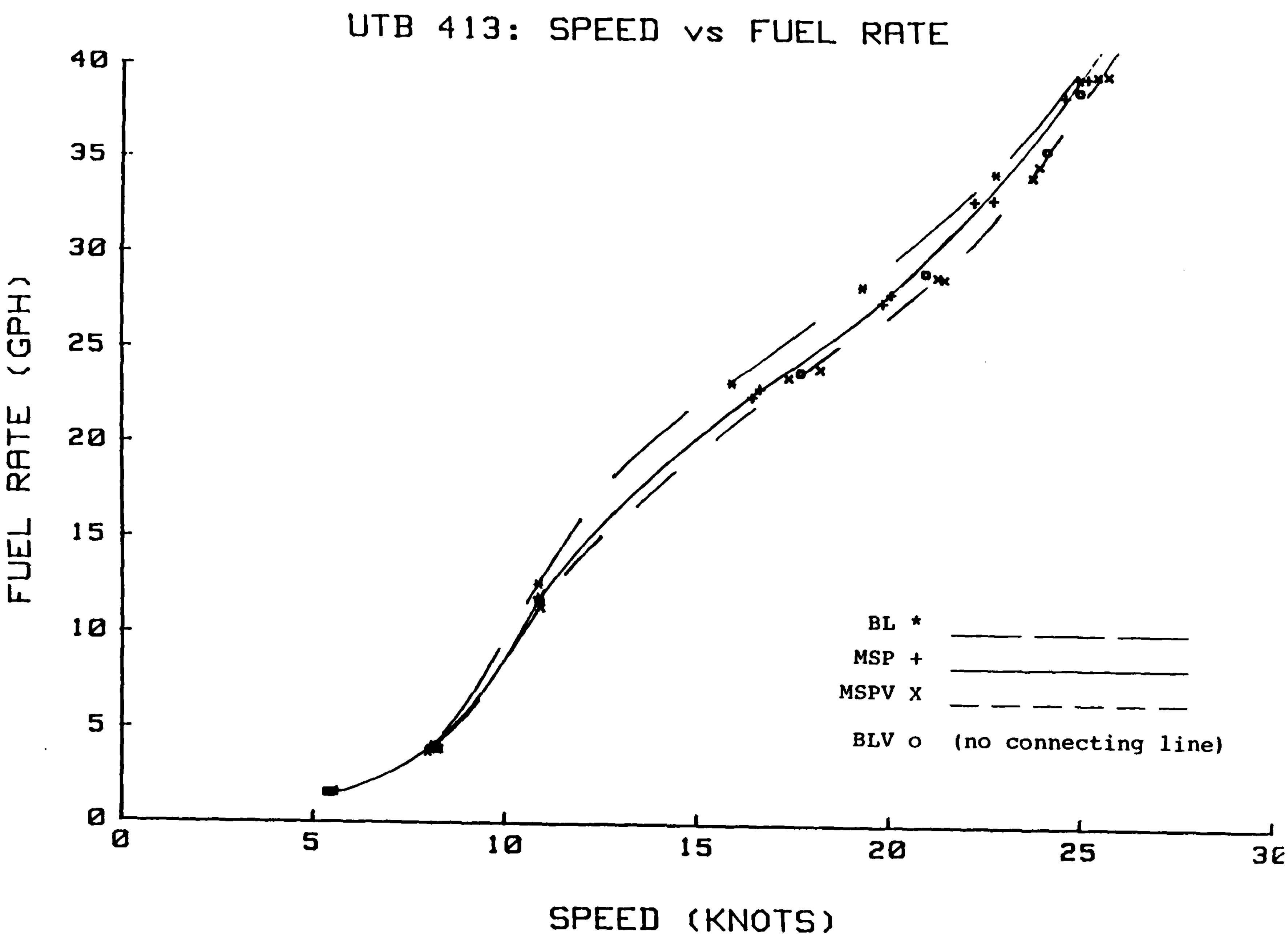

FIGURE 5-4: UTB 413 - SPEED vs FUEL RATE

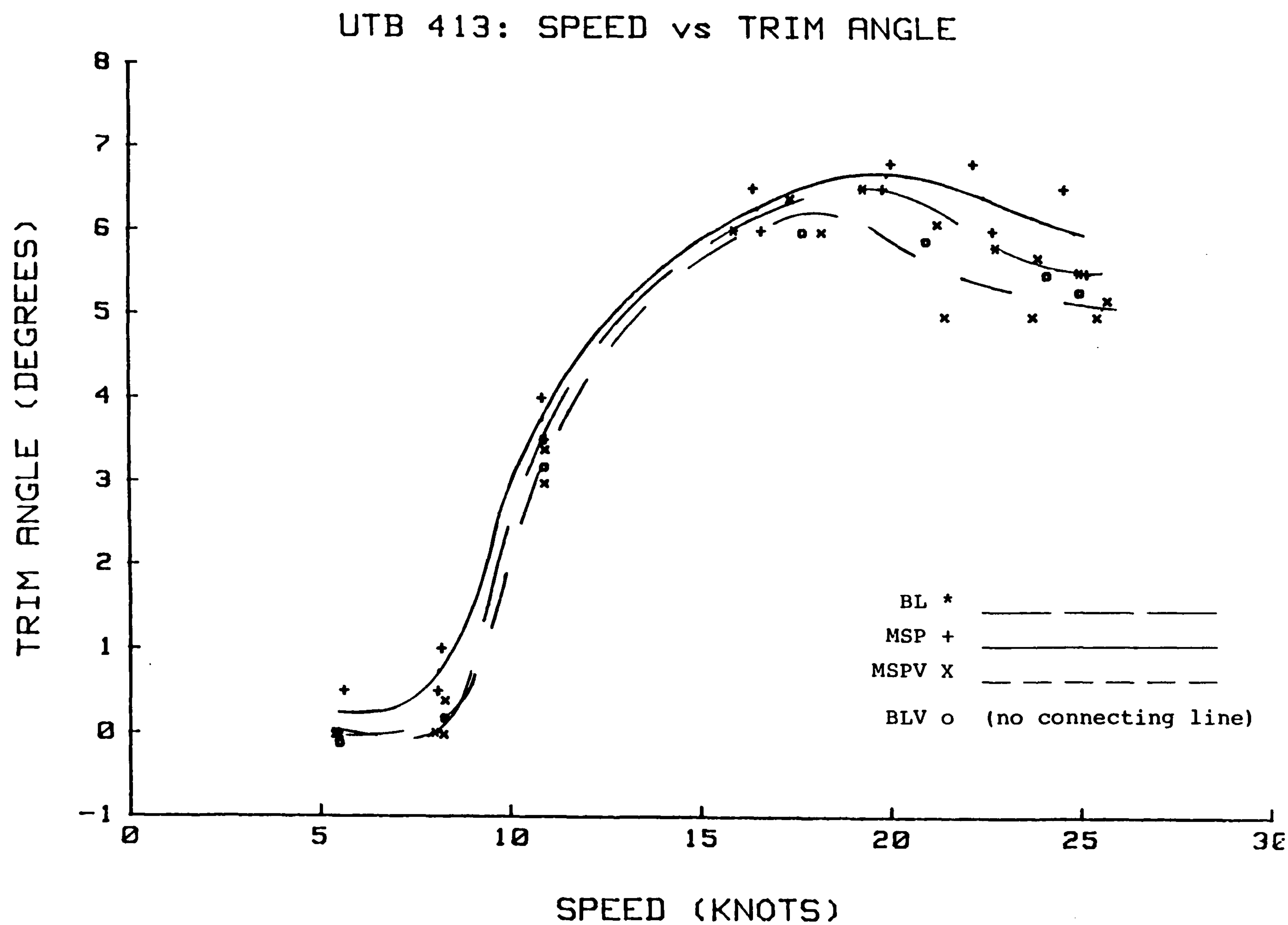

FIGURE 5-5: UTB 413 - SPEED vs TRIM ANGLE

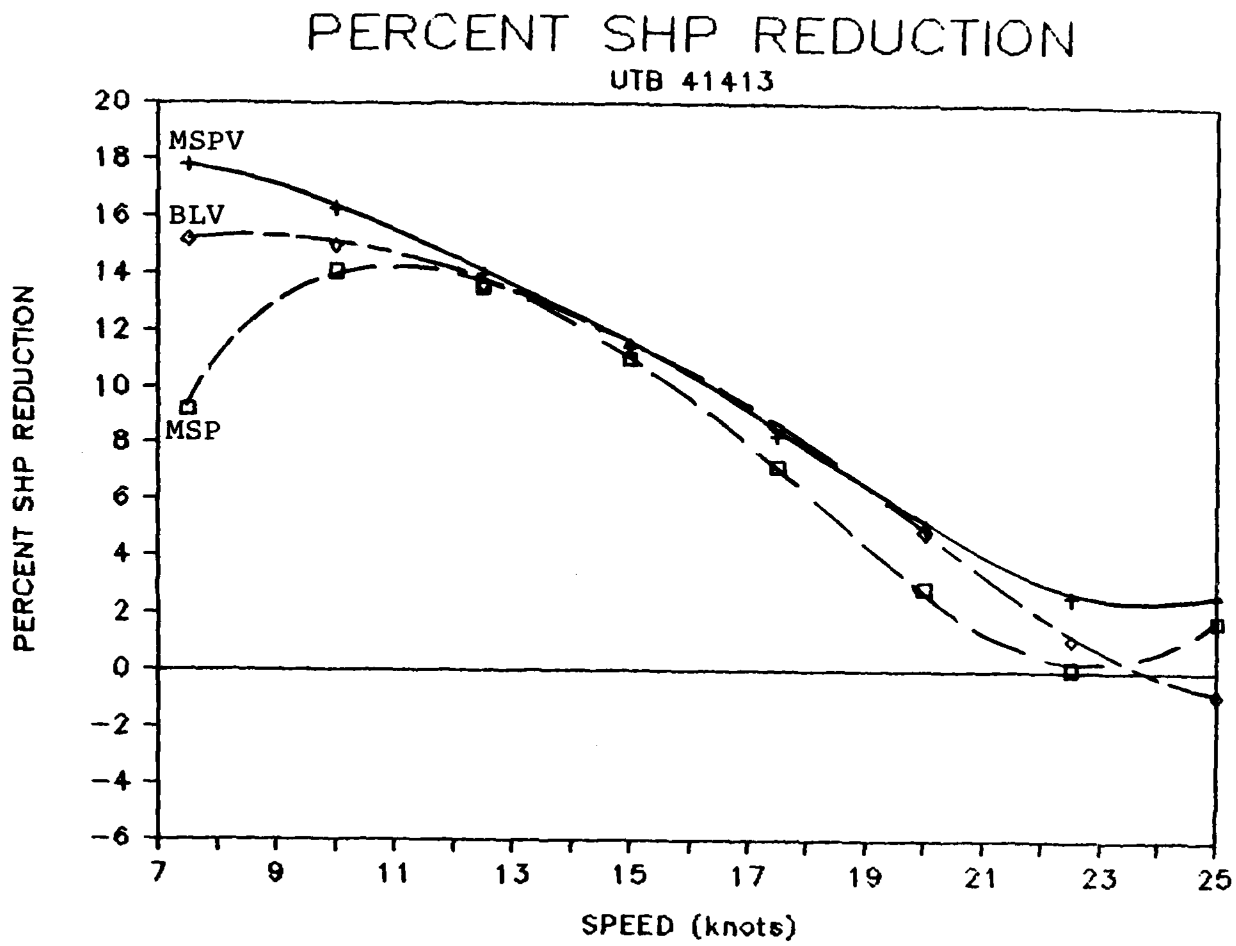

FIGURE 5-6: PERCENT SHP REDUCTION RELATIVE TO BASELINE PROPELLER

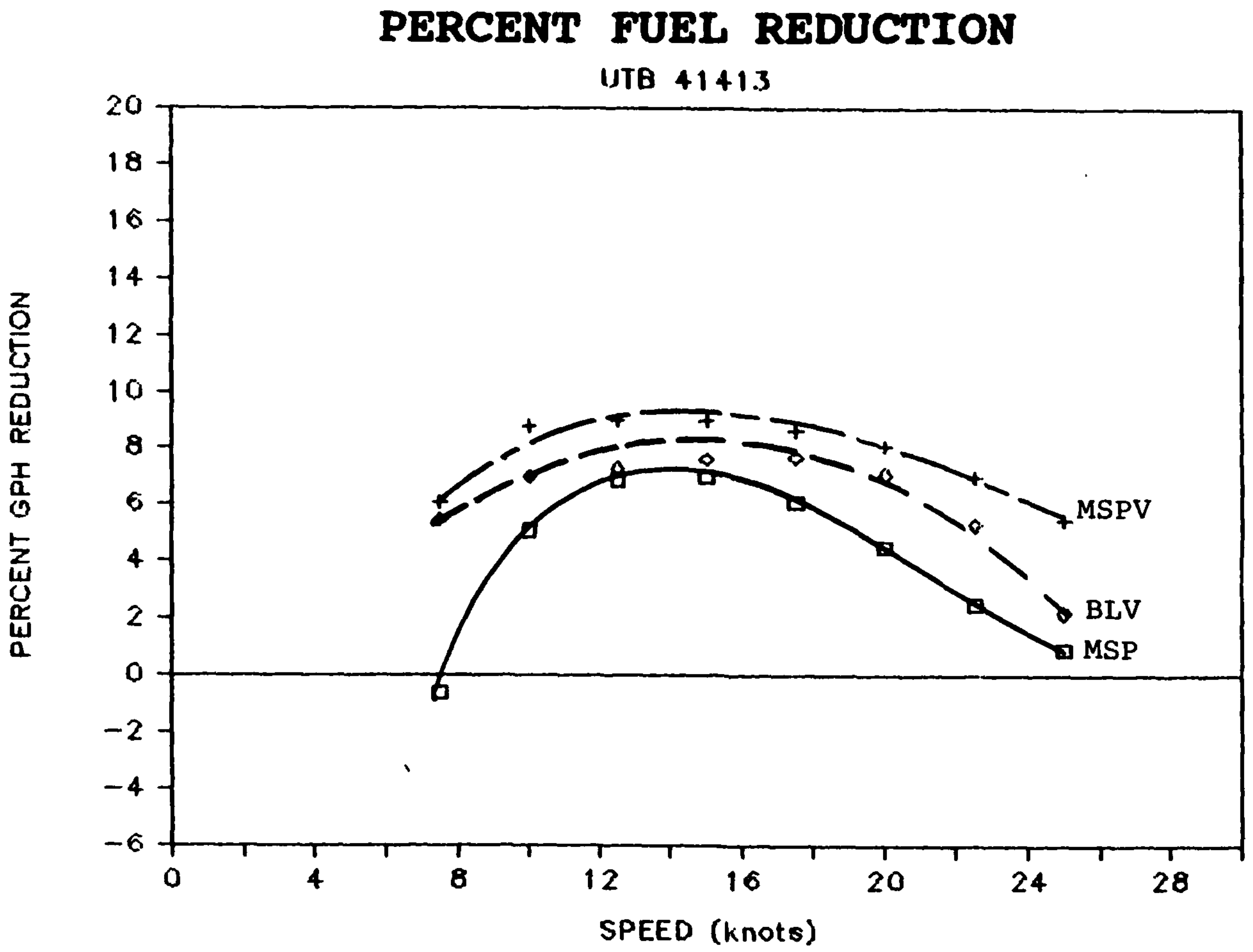

FIGURE 5-7: FUEL SAVINGS RELATIVE TO BASELINE PROPELLER (current results)

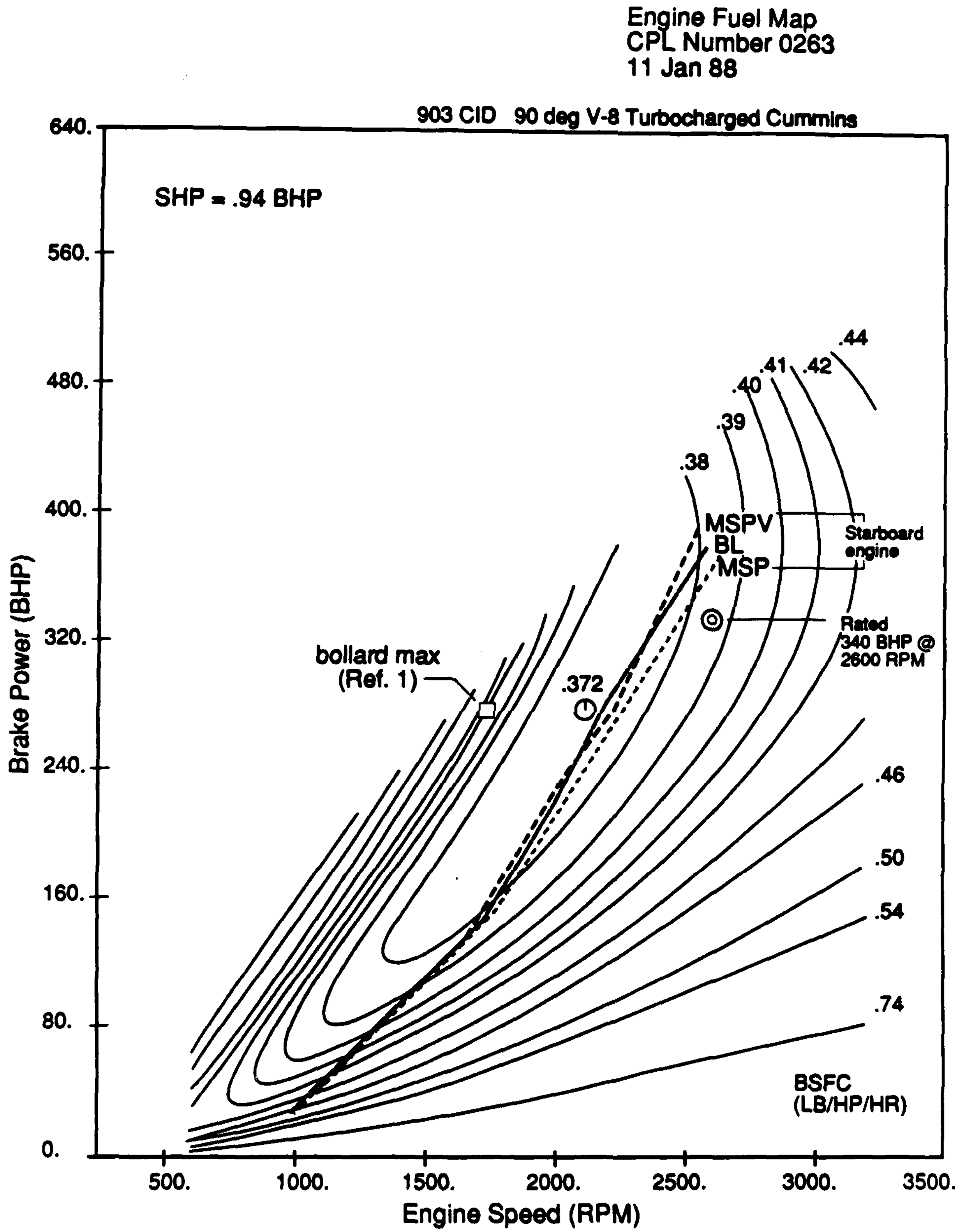

FIGURE 5-8. ENGINE FUEL MAP

from those used in the current tests. If one considers only the difference between MSP and MSPV, Figures 3-2 and 5-7 both indicate approximately 3-6% improvement at all speeds due to the vanes. Similarly, if one compares the BL with BLV data, both series of tests indicate higher savings with the vanes at low speeds (8-15%), declining to almost nothing at full speed. Thus, despite the apparent difference between Figures 3-2 and 5-7, both series of tests show that the vanes produce significant fuel savings.

Both series of tests also indicate that there may be a problem with all of the tested propulsors, as they exceed the recommended rated power levels, and do not achieve full rated RPM. Table 5-1 summarizes the performance of the starboard engine at the highest measured RPMs with the various propulsors.

TABLE 5-1
SUMMARY OF DEVELOPED HORSEPOWER AND RPM (STARBOARD)[4]

	BL	MSP	BLV	MSPV	RATED*
SHAFT RPM	1291	1299	1250	1274	1300
SHAFT HP	348	340	356	360	318

* Light Duty Continuous (1,000 hours/year)

All four configurations are seen to develop greater HP than the manufacturer's recommended light duty continuous rating, particularly the props with vanes. The data of references [1] and [9] also support this finding. This operating condition can significantly increase maintenance costs and reduce engine life. (All tests were performed on freshly cleaned and painted hulls, so the problem will become even worse as the bottom fouls.) The problem can be partly alleviated by lowering the fuel stops,

[4]Despite the nominal symmetry, the starboard engine consistently produced greater torque than the port at equal RPM. The excessive torque problem also exists on the port engine, but to a lesser extent.

or operating at slightly reduced throttle. This would enable operation at the rated HP, but at a considerably lower RPM than suggested by the manufacturer. A more complete solution would be to reduce either the diameter or the effective pitch of the propeller by approximately one inch[5].

It is difficult to justify these measures retroactively without more precise information on the costs of overloading the engine, but such a remedy should certainly be considered when ordering new props for these vessels. Note that the average boat only operates 78 hours/year at full speed (see Table 6-1), and would most likely not experience a maintenance problem.

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[5]Estimated from Troost B-series results.

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6.0 ECONOMIC ANALYSIS - NEW PROPS AND VANES

In this section, we will attempt to estimate the cost of installing the prop and vane set on 200 active 41 foot UTBs. Cost reduction for quantity purchasing will be included and estimates of the labor involved in the installation of the vanes will be made. Fuel savings will be allotted to three speed ranges (low, medium and high) in accordance with surveys of vessel operations.

Other benefits and liabilities, such as reduced plating vibration and increased maintenance due to higher horsepower operation, will not be included. It will be assumed that the same operating speed profile will be maintained, with a resulting reduction in fuel consumption.

6.1 Fuel Savings

The reduction in fuel consumption and the resulting savings are shown in Table 6-1. The operating profile was determined after several conversations with boat operators in various districts. The values of GPH saved are taken from fitted test data at the indicated speeds.

6.2 Maintenance

Some assumptions must be made concerning the maintenance of the vanes. The present manufacturing and assembly technique requires a lot of time and expense. Perhaps, in the future, the vane set could be cast in one piece and attached to the strut by mechanical devices, but for the purpose of this analysis, we must take the present configuration and evaluate it.

The vanes are presently manufactured in segments which are welded together on a hub. This is then welded to the existing strut that supports the outboard propeller bearing. In order to produce a good weld, all paint must be removed from the existing weld fillet between the strut and the bearing housing. The vane

TABLE 6-1

RECURRING ANNUAL FUEL SAVINGS

(NEW PROPS AND VANES)

Speed	Hours	GPH Saved	Gallons Saved
Low (0-7 knots)	474	.14	66
Medium (15 knots)	98	1.96	192
High (24 knots)	78	2.26	176
TOTAL	650	-	434

Estimated Fuel Price = $0.85 per gallon
Annual Fuel Saving per boat = $369
Total Annual Savings (200 boats) = $ 73,800

hub must be trimmed back so it will slide over the housing. Holes for the bearing set screws must be drilled in the vane hub to suit the existing bearings. The bearing must be removed because of the heat involved in the welding process, and the shaft must be pulled to get the bearing out. The whole process takes two days and we will use a nominal estimate of $1,000.00 cost for this to be done at a Coast Guard facility. It would probably be five times that much in a commercial yard.

The existing propellers cost about $900.00 each. In order to get a modified stock propeller into the system, we will assume its cost to be $1,000.00. The modified stock propellers used on 41309 and 41413 cost $4,600.00 each, but they were NI-BR-AL which is hard to repair and expensive. Even an order of 100 propellers would only get the cost down to $2,000 each. We, therefore, would recommend a standard bronze propeller and assume a cost of $1,000.00 each.

Propellers are replaced quite frequently on 41 foot UTBs. The data gathered from the Ship's Inventory Control Point in

Baltimore, MD, indicates that 45 propellers per year are added to the inventory to supply approximately 200 boats. No figures are available for propellers that are repaired and put back into the system. We will assume that approximately 20 vane units will have to be replaced in a given year, about half the number of propellers replaced. Propellers, however, can be changed in one hour and can be repaired by local services. The vanes have to be chiseled off and may not be repairable at all. The assumption will be made that 20 vane units per year will be removed, replaced and not repaired, as noted in Table 6-2.

TABLE 6-2

TOTAL FLEET-WIDE
RECURRING ANNUAL MAINTENANCE COSTS
(NEW PROPS AND VANES)

Differential Cost of Propellers ($100 ea. x 45)	=	$ 4,500
Removal of Damaged Vanes ($100 ea. x 20)	=	$ 2,000
Cost of Replacement Vanes ($2000 ea. x 20)	=	$40,000
Installation of Replacement Vanes ($300 ea. x 20) (Includes new bearings, etc.)	=	$ 6,000
	TOTAL	$52,500

6.3 Discounted Savings

Using the Economic Analysis Guidelines from NAVFAC Publication P-442, the values from Tables 6-1 and 6-2 are summarized in Table 6-3. It has been assumed that the life of the project would be 15 years, since many of the boats are already 15 years old. In Table 6-3, the Discount Factor is obtained from NAVFAC P-442 and is a reflection of the "cost of money" at an assumed rate of 10% per annum, so that the net present value can be calculated and compared to a present expenditure.

6.4 Funding Requirements

The funding requirements for this project are listed in Table 6-4. Once again it is assumed that the differential cost

TABLE 6-3

TOTAL FLEET-WIDE DIFFERENTIAL DISCOUNTED SAVINGS
(NEW PROPS AND VANES)

Years	Annual Savings (Table 1)	Annual Cost (Table 2)	Annual Diff. Savings	Discount Factor (1)	Total Net Present Value of All Differential Savings
15	$73,800	$52,500	$21,300	7.980	$169,974

(1) The Discount Factor takes into consideration the compounded cost of money.

for the modified stock propellers will be $100.00 each and that the batch price for a vane unit will be $2,000.00 as quoted by the present manufacturer for an order of 200 vane units. It is also assumed that the funding would be expended during one fiscal year.

TABLE 6-4

DIFFERENTIAL COST OF INITIAL VANE INSTALLATION

Cost of Vane Set (2 units per boat)	$4,000
Installation Cost (2 units per boat)	$ 300
Differential Cost of Propellers (2 per boat)	$ 200
	$4,500
Times No. of Boats	200
TOTAL COST	$ 900,000

$$\text{Savings/Investment Ratio} = \frac{\$169{,}974^{*}}{\$900{,}000} = 0.19$$

* from Table 6-3

6.5 Summary of Economic Analysis of Modified Stock Props and Vanes

For an investment to be economically attractive, the Savings/Investment Ratio (SIR) must be greater than one. If it is less than one, the present value savings are not sufficient to amortize the investment cost. Thus, the vane sets, in their current configuration, do not save enough fuel, at current prices, to offset the anticipated costs of installation and repair. If, however, the cost of manufacture, installation and repair can be reduced, or greater fuel savings achieved by better design methods, or the price of fuel increasing vane retrofits could still be attractive.

Since the differential costs of the modified stock propellers alone are much less than with vanes, the next section considers the economics of the modified stock propellers without vanes.

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7.0 ECONOMIC ANALYSIS - NEW PROPS ALONE

7.1 Fuel Savings

In this section, we will attempt to estimate the cost and savings resulting from the use of a different propeller on the 41 ft. UTB. This propeller has a variable pitch (25 in. at the hub to 30 in. at the tip) as opposed to the constant 26 in. pitch of the existing propeller. Table 7-1 summarizes the potential fuel savings achieved in a manner similar to Table 6-1.

TABLE 7-1
RECURRING ANNUAL FUEL SAVINGS
(NEW PROPS ALONE)

Speed	Hours	GPH Saved	Gallons Saved
Low (0-7 knots)	474	0	0
Medium (15 knots)	98	1.54	151
High (24 knots)	78	0.52	41
TOTAL	650	-	192

Estimated Fuel Price = $0.85 per gallon

Annual Fuel Saving per boat = $163

Total Annual Savings (200 boats) = $32,600

7.2 Maintenance

As mentioned in Section 6.2, the existing propellers cost about $900 apiece, and could be replaced by propellers costing about $1,000 apiece. We know that about 45 propellers per year are added to the inventory to supply 200 boats. What is not known, however, is the number of propellers that are repaired and reintroduced to the inventory. There may be a higher cost associated with the repair of the new variable pitch propellers as opposed to the constant pitch of the old ones; however, we will assume for the sake of this study that there will be no increase in repair cost. The additional cost of modified stock propellers versus old will therefore be the $100 differential

cost times the 45 replacement propellers, a total of $4500 per year.

7.3 Discounted Savings

Using the same technique described in Section 6.3, we will apply the discount factor to the difference between cost and savings in Table 7-2.

7.4 Funding Requirements

Again using only the differential cost of each propeller, $100, we can summarize the additional funding for 200 boats as $40,000 in Table 7-3.

TABLE 7-2

FLEET-WIDE DISCOUNTED DIFFERENTIAL COST AND SAVINGS
(NEW PROPS ONLY)

Years	Savings	Cost	Differential	Discount Factor	Discounted Savings
15	$32,600	$4,500	$28,100	7.980	$224,238

TABLE 7-3

SAVINGS/INVESTMENT RATIO

Differential Cost of Initial Propellers (2 per boat, 200 boats)	$ 40,000
Discounted Savings (from Table 7-2)	$224,238

Savings/Investment ratio = $\frac{\$224,238}{\$\ 40,000} = \underline{\underline{5.6}}$

7.5 Summary of Economic Analysis of New Props Alone

The Savings/Investment ratio of 5.6 indicates that the payback period would be less than one year for the installation of the new props alone. If the results of the one-year

operational evaluation indicate no cavitation problems, we will recommend the procurement of the modified stock propellers and their addition to the inventory, to be installed in pairs as the supply of old propellers is depleted. It is important to note that the modified stock propellers must be operated at a lower RPM, to attain the previous top speed without increasing the torque on the engine. Then the fuel savings will be realized without increasing maintenance costs on the engines.

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8.0 CONCLUSIONS

1. The Modified Stock Propellers (MSP's) (without vanes) reduce the HP and fuel requirements of the Baseline (BL) props over the entire speed range. Clearly the MSP's linearly increasing pitch distribution is superior to the BL's constant distribution. Whether this results from a more favorable radial distribution of circulation, or from a more nearly "wake-adapted" design, is uncertain at this time.

2. The addition of the vanes improves the performance of both the BL and MSP's, so that both vaned combinations are superior to the props without vanes, the Modified Stock Prop with Vanes (MSPV) combination being best of all.

3. The final ranking of the propulsors from worst to best is: BL, MSP, Baseline with Vanes (BLV), Modified Stock Propellers with Vanes (MSPV).

4. Both series of tests (1986 and 1988) show that the vanes produce significant fuel savings. Compared with the baseline (BL) propellers alone, the addition of vanes produces fuel savings of 8-15% at low speeds, but the percentage reduction at full speed is quite small. When the vanes are installed with the modified stock propellers (MSP) an improvement of 3-6% is realized over the entire speed range, compared with the MSPs alone. In addition, the modified stock propellers' fuel consumption seems substantially better than that of the baseline propellers, but it remains unclear whether this is due to the difference in pitch distributions or to the poor matching of the BL propellers.

5. Application of the vanes to both the BL and MSP's results in a reduction in RPM at equal boat speed. The vanes also appear to increase torque at the high speed, while decreasing it at the middle and low speeds. In all cases, the changes

are relatively small, so that retro-fitting a vane set will not significantly change the operating point of a propeller which has been selected to conform to rated power and RPM.

6. At maximum throttle setting, all of the propulsors tested operated at more than the manufacturer's rated power, at RPMs which were either equal to or lower than the rated value. It is suggested that the throttles normally be set to produce the rated power, although higher power operation may be permissible for short periods of emergency operations. The RPMs in these cases, however, would be well below the manufacturer's rating. The possibility of using reduced propeller diameter or reduced pitch should also be considered, so that rated power can be achieved at rated RPM.

7. There is no reason to believe that any of the propulsors tested represent an optimum design. Greater fuel and HP savings may well be possible with the development and verification of improved design procedures. Reduced circumferential variation of the inflow to the propeller should enable a reduction in required blade area, and permit additional savings.

8. While there was no apparent cavitation on any of the propulsors tested, both the MSP and MSPV's are currently undergoing a one year operational evaluation, and will be inspected for cavitation and other structural damage during the year. Any mismatch between the pitch distributions of the propeller and vanes will prevent the blade sections from operating at their ideal angle of attack, and increase their tendency to cavitate.

9. These tests show that significant fuel savings can be achieved, both by more attention to conventional propeller design methods, and by the use of pre-swirl vanes.

10. The MSPs would cost only slightly more than the present BL propellers, and economic analysis shows that they are cost-effective retrofits for the 41′ utility boats. The vanes, however, add substantially to the initial and maintenance costs of the propulsive system. Economic analysis shows that these costs cannot be recovered on the 41′ utility boats, because their present low speed, light duty operations do not consume very much fuel in the first place. For vessels with more frequent, higher speed operating profiles, however, pre-swirl vanes might offer significant cost savings. Increased fuel prices, improved methods of vane fastening and removal, and improved design methods would further increase the attractiveness of pre-swirl vanes.

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9.0 RECOMMENDATIONS

1. While the pre-swirl vane concept shows definite promise, a poor design also has the potential for serious cavitation problems and engine operation at non-recommended conditions. Further study and verification of existing design methods is therefore strongly recommended.

2. A successful design method must have an accurate knowledge of the inflow velocities to the propeller. A full-scale wake survey of a self-propelled 41′ UTB would provide valuable research data and should be seriously considered.

3. The pre-swirl vane concept may show even greater promise in cases where the propeller loading is higher due to: high thrust, low diameter and/or low speed of advance. Typical cases include icebreakers, mine-sweepers and towing vessels. Further research into this possibility is recommended.

4. It does not appear economically worthwhile to retro-fit the present vane design by present methods on a fleet-wide basis. The Modified Stock Propellers (MSPs) alone, however (perhaps with slightly reduced pitch or diameter), should definitely be purchased to phase out the old baseline propellers on all 41′ UTB's.

5. Additional research on radial load distributions is recommended to help understand why the MSP performs so much better than the BL propeller.

6. The CG 41′ UTB operators should calibrate their onboard tachometers with a hand-held strobe to ensure that the propeller RPMs called for from the bridge are actually obtained.

7. More attention should be given on twin screw vessels to precise matching of the port and starboard propeller weights. Stock propellers have been found to vary in weight as much as 34%.

REFERENCES

1. G. Larimer et al., "Reaction Fin Applications in Marine Propulsion", SNAME Hampton Roads Section Meeting, March 5, 1987.

2. H. Glauert, "Airplane Propellers", Chapter 3 of Aerodynamic Theory, edited by W. Durand, California Inst. of Technology, 1943.

3. H. Lerbs, "Moderately Loaded Propellers with a Finite Number of Blades and an Arbitrary Distribution of Circulation". SNAME Transactions, 1952.

4. M. Eckhardt, W. Morgan, "A Propeller Design Method". SNAME Transactions, 1955.

5. W. Gearhart, "Advanced Propulsor Design Studies", U.S. Coast Guard Report CG-D-1-86, June 1985.

6. T. Brockett and R. Corpus, "Marine Propulsors for Minimum Shaft Horsepower", 21st ITTC Proceedings, 1987.

7. J. Kerwin, W. Coney and C. Hsin, "Optimum Circulation Distributions for Single and Multi-component Propulsors", 21st ITTC Proceedings, 1987.

8. L. Bowling, "Design of a Non-axisymmetric Stator for Improved Propeller Performance", MIT Ocean Engineering Dept. Report 87-10, June 1987.

9. C. Shields, "Small Craft Performance Characteristics With and Without a Reaction Fin Propulsion System", NAVSEACOMBATSYSENGSTA Report No. 60-184, October 1987.

10. F. Todd, "Resistance and Propulsion" Chapter VII of Principles of Naval Architecture, edited by J. Comstock, SNAME 1967.

11. K. Christopher, "Effect of Shallow Water on the Hydrodynamic Characteristics of a Flat-Bottom Planing Surface," NACA Technical Note 3642.

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APPENDIX A

INSTRUMENTATION DESCRIPTION

For this test series fuel consumption was the primary measurement of interest. However, several other variables were measured, as shown on the list below:

Device	Variables Measured
Fuel Meter	flow rate, total fuel consumed
Torque Meter*	shaft rpm, torque
Inclinometer	trim
Thermocouples	fuel temperature
Accelerometers	hull acceleration/vibration

*used on test boat 41413 only

A schematic of the instrumentation layout is given in Figure A-1. It is the purpose of this appendix to discuss the basic principles behind each device and describe the calibration process, where applicable.

Fuel Meter

The system used was a Fluidyne measurement system designed for precision determination of diesel fuel consumption of light

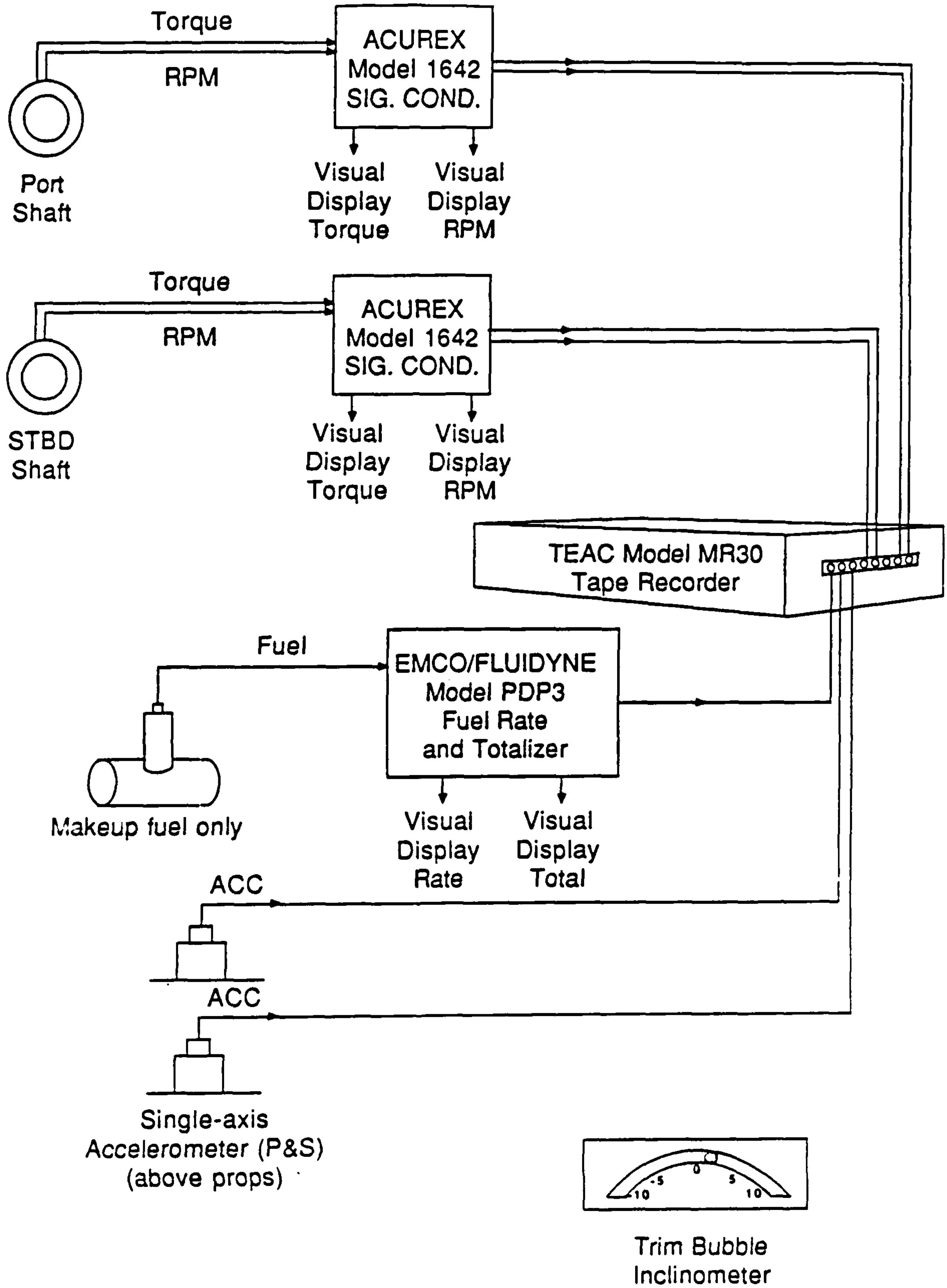

FIGURE A-1 INSTRUMENTATION DIAGRAM

to heavy-duty diesel vehicles of up to 2,000 HP. The fuel measurement system consists of a PDP3 four piston positive-displacement flow rate transducer capable of measuring from 0.16 to 95.1 gph with an accuracy of 0.5% and repeatability of 0.1%. The transducer is connected to a digital indicator which displays instantaneous flow rate and total fuel consumed. The system was also modified to provide a continuous analog signal of the flow rate which was then recorded.

The installation of the fuel measurement system in the 41' UTB was complicated by the fuel injection system on the engines. This system delivers excess fuel to the injectors, returning the unused fuel to the tanks. This excess fuel is used to cool the injectors. The fuel injection system will bypass many times more fuel than the engines use. The technique of utilizing four flow meters, two on the supply lines and two on the return lines, and subtracting the latter from the former, will not provide accurate measurement since metering errors are additive. Also, the return fuel is hot and aerated, leading to further measurement error.

An effective technique for dealing with return fuel is to reroute the return fuel line, normally terminated at the fuel tank, through a heat exchanger and to a vented collecting tank (see Figure A-2). This collecting tank has a constant-level float which calls for make-up fuel as required. The make-up fuel comes from the main tanks and is measured by the flow meter

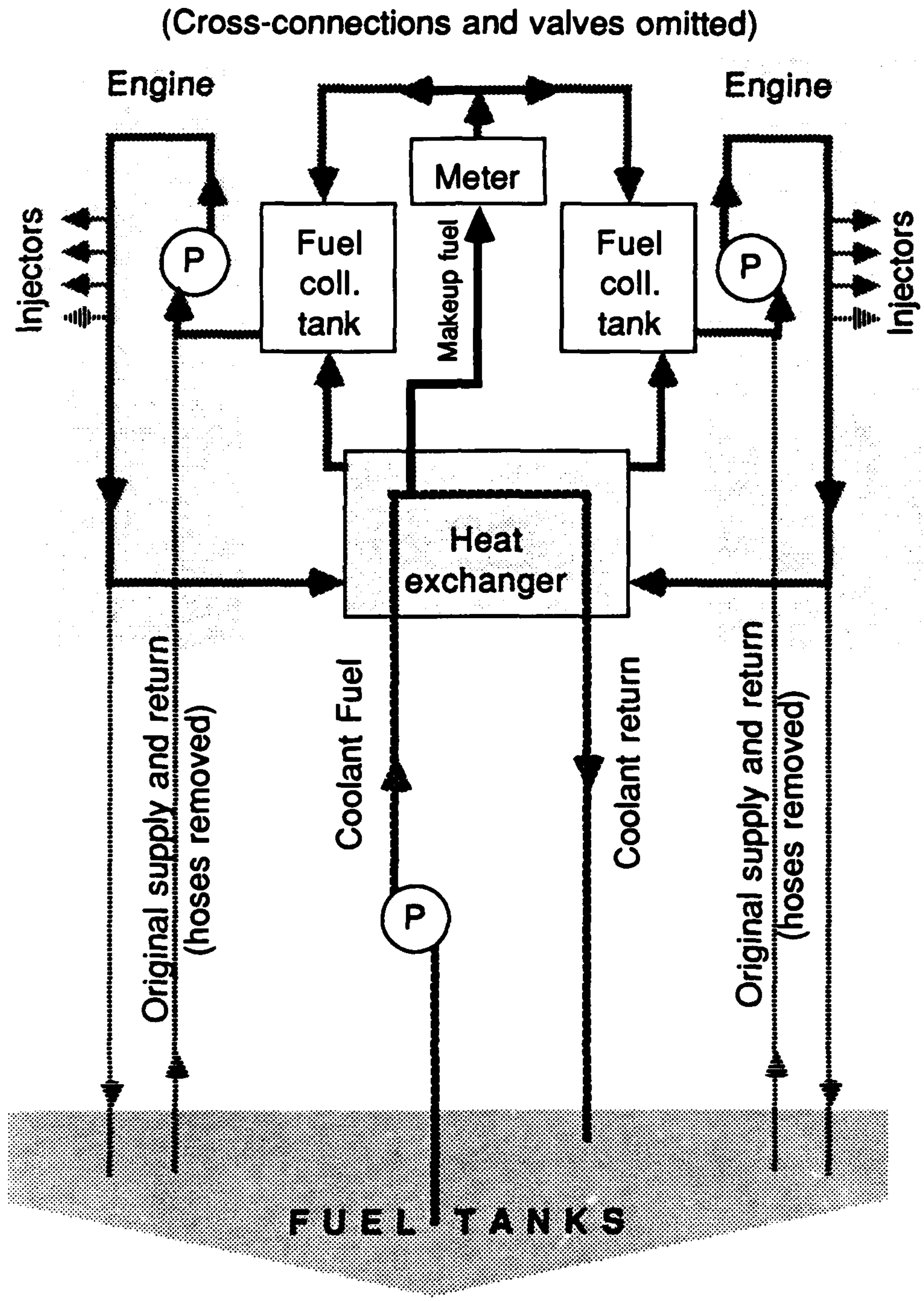

FIGURE A-2 Schematic Diagram of Fuel System
41' UTB Testing, R&DC, Spring 1988

as it is added to the recirculating fuel. The heat exchanger coolant is also provided from the main tanks and is returned to the tanks for heat exchange by the hull.

For the purposes of this test program, the transducer, collecting tanks, heat exchanger and auxiliary pump were mounted on a common platform. This made installation and removal relatively straightforward. Photos of the installation are shown in Figure A-3.

The fuel meter is bench calibrated to an accuracy of 0.5% with repeatability of 0.1%. When calibrated at sea as installed in the vessel, we could verify only an accuracy of 1% with use of a day tank and hand measurement of fuel volumes. However, repeatability is the key to comparative measurements such as baseline props versus props and vanes. To assure repeatability of the measurements, long run-in periods (1 to 2 minutes) were used to stabilize the fuel rate before entering the test course.

Torque Meter

The system used was an Acurex Universal Power Measurement System whose purpose is to obtain torque and rpm data from high or low speed rotating shafts. For our tests, we had two such systems so we could measure the torque and rpm on each shaft simultaneously.

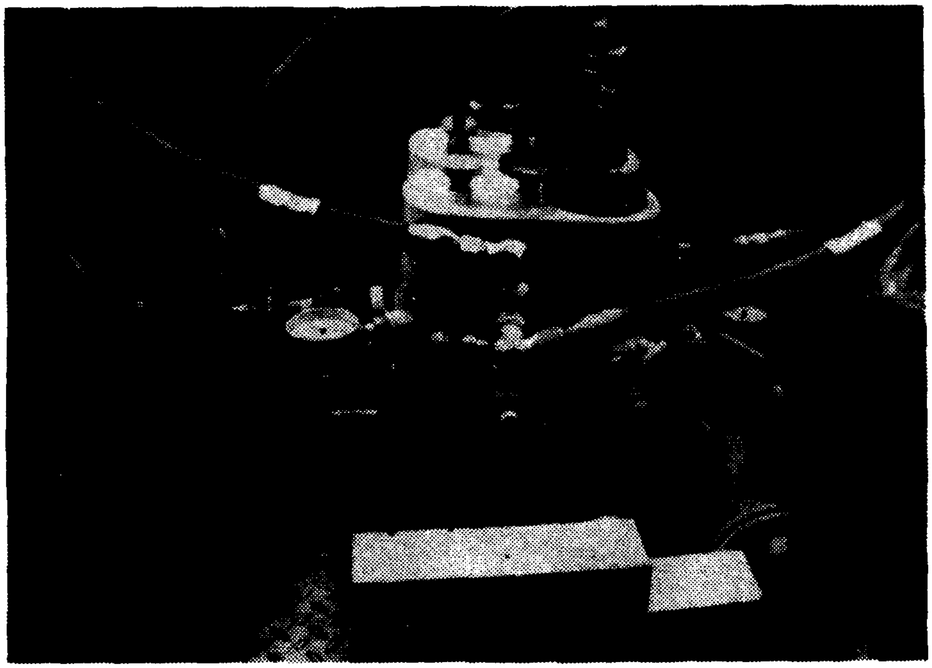

FIGURE A-3

FUEL METER INSTALLED ON 41' UTB

The power measurement system consists of a clamp on collar which fits closely around the propeller shaft. The collar has a built-in antenna, metal tabs for shaft speed detection and a pocket to hold the transmitter module. An electronics unit provides induction power to a stationary power separation unit with a loop antenna fitted around, but not touching, the clamp-on collar. The loop antenna receives the transmitted RPM and torque signals. The torque signal, along with the shaft speed signal, is routed to the electronics package. The readout is set up to display torque and rpm in engineering units. The set-up is shown on Figure A-4.

The clamped collar and transmitter module together rotate with the shaft and power a four-arm strain gage bridge. They also transmit the output signal to the non-rotating antenna. For our tests, strain gauge rosettes were epoxied to the shafts between the reduction gear coupling and the stern-tube bearing.

In order to calibrate the power meters, known moments were applied to the shafts, and correlated with the strain gauge output. To accomplish this, a 7-foot lever arm was fashioned from a 4-in x 4-in square aluminum channel. Steel brackets were bolted to one end of the arm, bored to accommodate the standard propeller taper and fitted with a keyway. Thus the arm was attached to the shaft in the same way that a propeller hub is usually fitted.

Inside the vessel, another bracket was bolted to the shaft

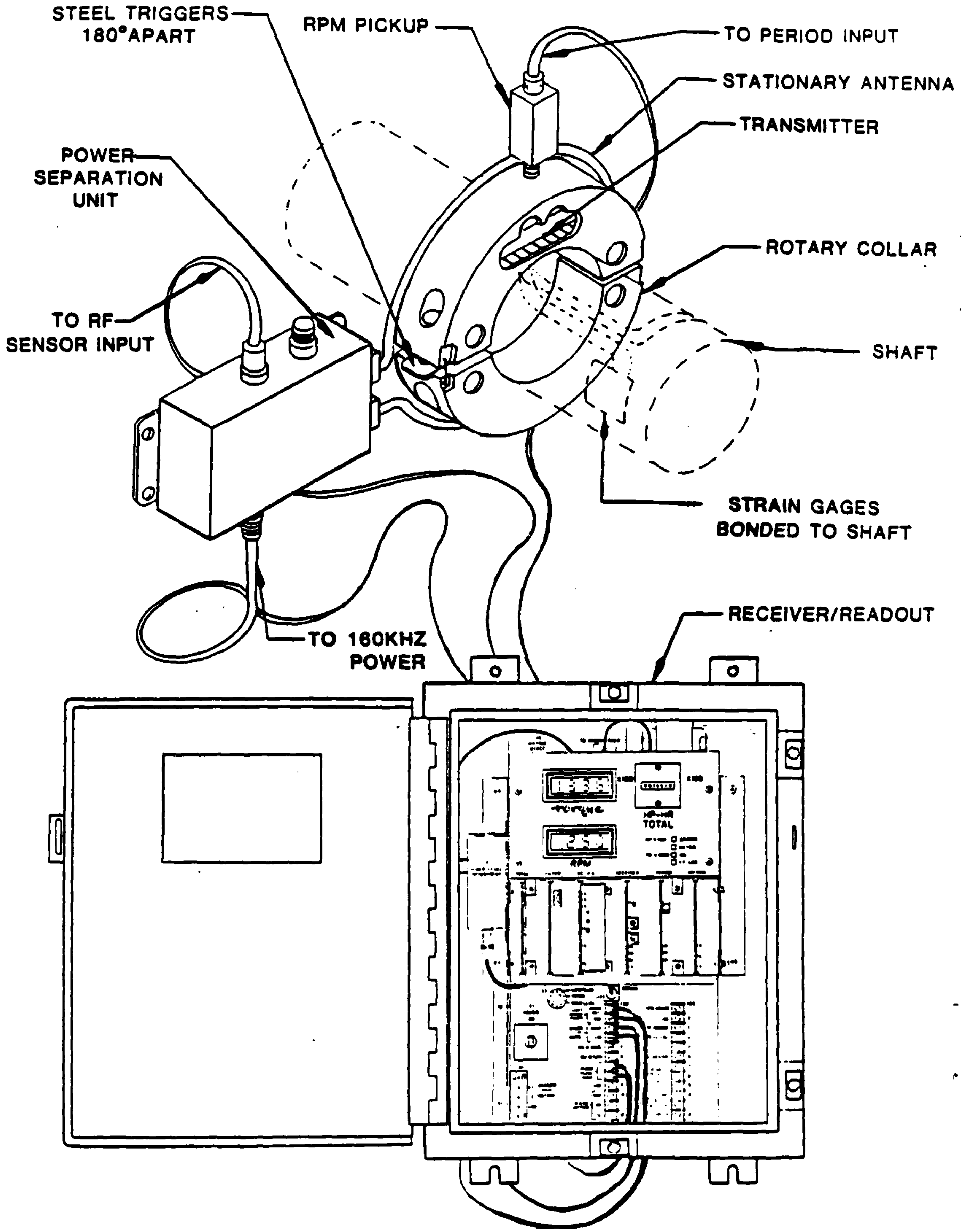

FIGURE A-4 POWER MEASUREMENT SYSTEM

using the existing holes for coupling to the reduction gear. This bracket braced the shaft against a nearby transverse frame and prevented the shaft from rotating. A series of weights were then suspended from the outboard end of the arm to provide known moments. This is shown in Figure A-5. These moments were corrected for the inclination of the shaft and the arm, and the moment produced by the arm alone was added to obtain the final quantity. Then the digital meter and tape recorded signal were calibrated against known moments. There was some concern about the effects of bearing friction, but the shafts turned freely by hand and the signals returned to zero when the arm was unloaded. There was also concern about the effect of bending moment on the torque signals during calibration, but laboratory tests showed that this effect was negligible. The arm calibrations were linear and passed within a few lb-ft of the origin. It was not considered necessary, or prudent, to apply the maximum expected full scale moment of 1500 lb-ft. Shaft stiffness moduli were inferred from these measurements and were 3-13% higher than the published for Aquamet steel. Therefore, the calibration procedure provided considerably greater accuracy than could have been obtained by assuming a shear modulus.

The RPM reading was verified by comparing the meter readout with that of a hand-held tachometer held at the shaft itself. During the test, the meter readout was used by the coxswain to match the RPMs of the two shafts. The torque and RPM were recorded during the test run and the torque was later corrected

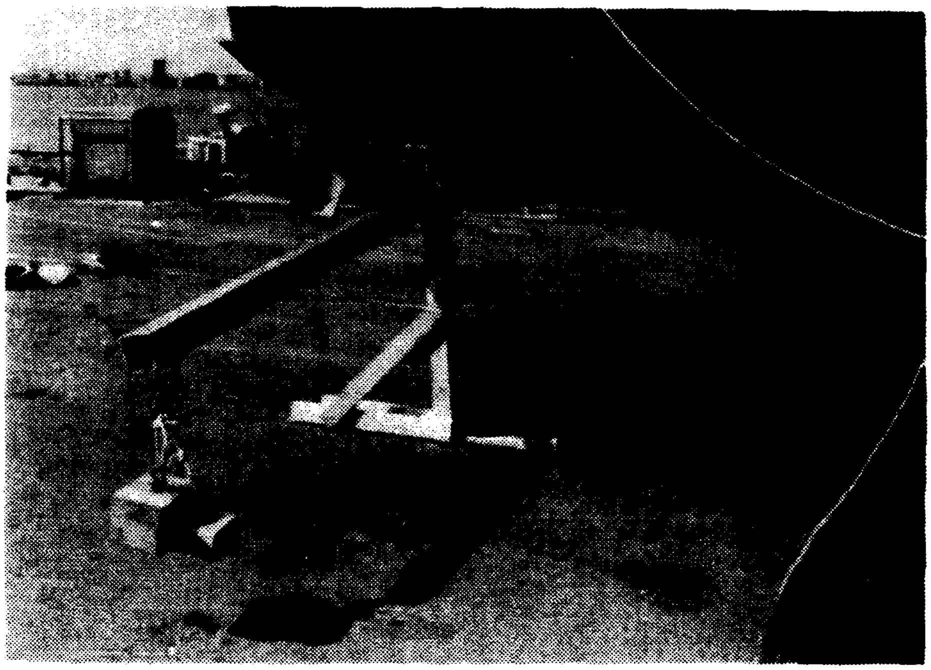

FIGURE A-5

ONBOARD CALIBRATION OF TORQUE METERS

in accordance with the calibration factor determined above.

Inclinometer

There were two types of inclinometers used during the tests. The first type was a bubble inclinometer, mounted inside the deckhouse next to the recorder. This inclinometer was read once prior to leaving the dock, then during each test run. It was accurate to 0.5 degrees. For the final series of tests it was desired to get instantaneous values of trim. For this we used an electronic inclinometer which had both a digital display and an analog signal suitable for recording.

Thermocouples

In order to determine the temperature of the fuel at various locations within the engine compartment, four thermocouples were installed; one on each collecting tank inlet and one on each fuel filter. Although the thermocouples were mounted externally, the fuel temperature was estimated to be approximately the same as the internal temperature. This gave an indication of the effectiveness of the heat exchanger. We also used the thermocouples in conjunction with the thermometers on the make-up fuel tanks to make sure the fuel was cooled sufficiently to begin the next run.

Accelerometers

Variations in hull vibration due to the modified stock prop and the vanes were measured by mounting accelerometers on the plating over the propellers. Vertical-axis piezoelectric accelerometers were used (B&K) with portable charge amplifiers. The output was recorded and analyzed and the results are summarized below in Table A-1. Generally, the vibrations increased slightly when the new propellers were tested, (perhaps due to their greater tip loading), and decreased when the vanes were added to that configuration. The data from the 2200 Engine RPM tests are shown. Results were similar at other RPMs.

TABLE A-1

MAXIMUM VERTICAL ACCELERATION OF PLATING - 2200 ENGINE RPM

Baseline Props	Modified Stock Props	MSP with Vanes
0.292 g	0.631 g	0.419 g

Recording

The fuel rate, port and starboard torque and shaft RPM, and hull plating acceleration were recorded on a Teac 7-channel cassette instrumentation recorder. This unit was recalibrated prior to each test.

APPENDIX B

DESCRIPTION OF BL, MSP AND VANES

The BL propeller has a diameter of 26 in., a constant pitch of 28" and a developed area ratio of 0.73. Additional data is given in Table B-1.

The modified stock propeller is identical to the BL in every way except for its pitch distribution, which is given in Table B-2.

The circumferential location of the vanes is shown in Figure B-1. The pitch distribution of vanes 1-4 is given in Tables B-3 and B-4. The existing strut was also converted into a vane by attaching a small trailing edge flap. In profile, the vanes have a swept-back leading edge, as shown in Figure 2-5, to shed debris and resist fouling.

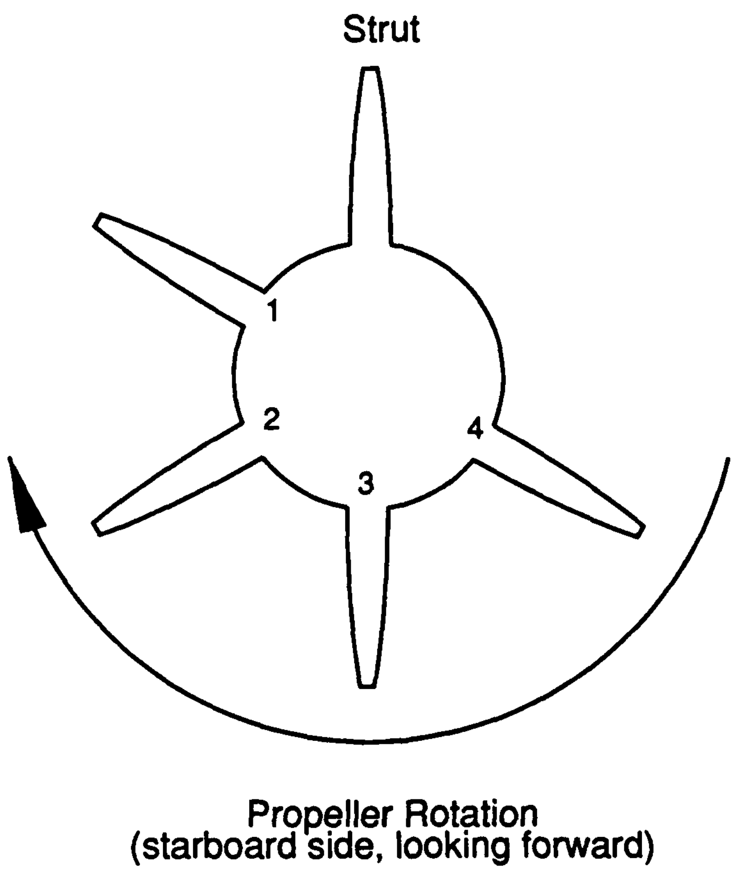

FIGURE B-1 Vane Geometry

TABLE B-1

Nondimensional rake, skew, chord, camber and thickness as a function of radius for both the baseline and modified stock propellers.

radius r/R	rake rake/D	skew *degrees*	chord c/D	camber f_0/c	thickness t/D
0.123	-0.0173	0.00	0.1239	0.0961	0.0346
0.200	-0.0161	-4.48	0.1731	0.0789	0.0321
0.300	-0.0144	-5.38	0.2349	0.0598	0.0288
0.400	-0.0128	-4.65	0.2927	0.0444	0.0255
0.500	-0.0111	-3.56	0.3442	0.0327	0.0222
0.600	-0.0095	-2.24	0.3852	0.0245	0.0189
0.700	-0.0079	-0.20	0.4063	0.0194	0.0157
0.800	-0.0062	2.81	0.3898	0.0162	0.0124
0.900	-0.0046	7.01	0.3128	0.0138	0.0091
0.950	-0.0037	9.63	0.2318	0.0127	0.0074
1.000	-0.0029	12.63	0.0000	0.0113	0.0058

TABLE B-2

Radial distribution of pitch for the modified stock propeller

radius	pitch	
r/R	*inches*	P/D
0.123	23.69	0.9112
0.200	24.63	0.9473
0.300	25.55	0.9825
0.400	26.21	1.0080
0.500	26.72	1.0277
0.600	27.19	1.0457
0.700	27.71	1.0658
0.800	28.39	1.0921
0.900	29.34	1.1285
0.950	29.94	1.1517
1.000	30.65	1.1790

TABLE B-3

PRE-SWIRL VANE CHARACTERISTICS
diameter = 20.0 inches
blades 1 and 2
(see Figure B-1)

r/R	CHORD/DIA.	THICKNESS/DIA.	CAMBER/DIA.	PITCH ANGLE
0.25	0.515	0.039	0.036	88.0
0.49	0.377	0.033	0.024	88.7
0.73	0.241	0.026	0.017	89.3
0.96	0.106	0.016	0.008	90.0

NACA 65 Mean Line
NACA 65 Thickness Distribution

TABLE B-4

PRE-SWIRL VANE CHARACTERISTICS
diameter = 20.0 inches
blades 3 and 4
(see Figure B-1)

r/R	CHORD/DIA.	THICKNESS/DIA.	CAMBER/DIA.	PITCH ANGLE
0.25	0.515	0.039	0.022	85.0
0.49	0.377	0.033	0.016	85.0
0.73	0.241	0.026	0.013	85.0
0.96	0.106	0.016	0.006	85.0

NACA 65 Mean Line
NACA 65 Basic Thickness Distribution

APPENDIX C

TABULATED EXPERIMENTAL DATA

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Test 107: **MODIFIED STOCK PROPS ALONE**

Run No.	Boat Speed (kts)	Fuel Rate (gph)	Port RPM (rpm)	Port Torque (lb-ft)	Port SHP (hp)	Stbd RPM (rpm)	Stbd Torque (lb-ft)	Stbd SHP (hp)	Trim Angle (deg)
1	5.62	1.60	299	88	5	300	107	6	.5
2	8.19	4.08	506	280	27	506	306	29	1.0
3	10.85	11.89	758	711	103	759	748	108	4.0
4	16.41	22.57	989	1064	200	1000	1109	211	6.5
5	20.03	28.06	1102	1165	244	1108	1234	260	6.8
6	22.19	32.96	1183	1242	280	1191	1310	297	6.8
7	24.57	38.57	1298	1320	326	1299	1375	340	6.5

Test 108: **MODIFIED STOCK PROPS ALONE**

Run No.	Boat Speed (kts)	Fuel Rate (gph)	Port RPM (rpm)	Port Torque (lb-ft)	Port SHP (hp)	Stbd RPM (rpm)	Stbd Torque (lb-ft)	Stbd SHP (hp)	Trim Angle (deg)
1	5.46	1.55	301	88	5	301	100	6	0
2	8.09	4.09	506	280	27	506	307	30	0.5
3	10.90	11.76	754	700	101	761	736	107	3.5
4	16.61	23.01	1004	1096	209	998	1077	205	6.0
5	19.81	27.59	1092	1160	241	1104	1216	256	6.5
6	22.69	33.03	1194	1231	280	1209	1320	304	6.0
7	25.16	39.57	1314	1325	331	1315	1355	339	5.5

Test 109: BASELINE PROPS ALONE

Run No.	Boat Speed (kts)	Fuel Rate (gph)	Port RPM (rpm)	Port Torque (lb-ft)	Port SHP (hp)	Stbd RPM (rpm)	Stbd Torque (lb-ft)	Stbd SHP (hp)	Trim Angle (deg)
1	5.39	1.57	300	88	5	301	103	6	0
2	8.01	3.77	490	257	24	496	295	28	0
3	10.87	12.60	785	870	130	778	800	118	3.5
4	15.87	23.30	1006	1138	218	994	1160	219	6.0
5	19.27	28.42	1097	1111	232	1093	1292	269	6.5
6	22.76	34.41	1213	1195	276	1218	1386	322	5.8
7	24.93	39.54	1304	1325	329	1291	1418	348	5.5

Test 110: MODIFIED STOCK PROPS AND VANES

Run No.	Boat Speed (kts)	Fuel Rate (gph)	Port RPM (rpm)	Port Torque (lb-ft)	Port SHP (hp)	Stbd RPM (rpm)	Stbd Torque (lb-ft)	Stbd SHP (hp)	Trim Angle (deg)
1	5.40	1.67	297	97	5	303	99	6	0
2	8.25	4.01	496	297	28	497	284	27	0
3	10.92	11.78	748	732	104	749	756	108	3
4	18.21	24.16	1000	1102	210	1016	1200	232	6
5	21.43	28.97	1107	1236	260	1103	1254	263	5
6	23.74	34.37	1194	1343	305	1192	1377	313	5
7	25.44	39.81	1275	1427	346	1274	1484	360	5

Test 111: MODIFIED STOCK PROPS AND VANES

Run No.	Boat Speed (kts)	Fuel Rate (gph)	Port RPM (rpm)	Port Torque (lb-ft)	Port SHP (hp)	Stbd RPM (rpm)	Stbd Torque (lb-ft)	Stbd SHP (hp)	Trim Angle (deg)
1	5.48	1.67	299	100	6	299	96	5	0
2	8.30	4.03	495	295	28	500	280	27	.4
3	10.93	11.49	745	744	106	749	724	103	3.4
4	17.39	23.70	991	1133	214	992	1168	221	6.4
5	21.25	29.06	1098	1232	257	1104	1295	272	6.1
6	23.89	34.99	1195	1334	303	1201	1404	321	5.7
7	25.72	39.85	1278	1415	344	1277	1474	358	5.2

Test 112: BASELINE PROPS AND VANES

Run No.	Boat Speed (kts)	Fuel Rate (gph)	Port RPM (rpm)	Port Torque (lb-ft)	Port SHP (hp)	Stbd RPM (rpm)	Stbd Torque (lb-ft)	Stbd SHP (hp)	Trim Angle (deg)
1	5.51	1.65	298	94	5	302	95	5	-.1
2	8.28	4.08	502	279	27	507	314	30	.2
3	10.91	11.78	756	697	100	753	775	111	3.2
4	17.70	24.01	1001	1061	202	1000	1220	232	6.0
5	20.95	29.29	1099	1189	249	1098	1340	280	5.9
6	24.11	35.83	1204	1313	301	1213	1469	339	5.5
7	24.97	38.99	1256	1397	334	1250	1495	356	5.3

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APPENDIX D

PARTIAL LOAD AND TRIM TAB EXPERIMENTS

Since many UTB's operate on relatively short SAR missions, well below their endurance range, it was desirable to see what fuel savings could be obtained by keeping the tanks half full, rather than continually "topping-off" the tanks. A comparison of half-load and full-load fuel rate data is in Table D-1 and plotted in Figure D-1. Small, but measurable savings in fuel consumption can result if one is willing to sacrifice the endurance reserve.

Tests were also conducted to investigate the effect of trim on boat speed and fuel consumption. Triangular wedges, each 12 inches long, 3 inches in width, and having a 15° slope were bolted to the underside of the transom lip. In the first series of tests, eight wedges were used. Significant trim reduction was achieved, but the fuel rate at any given speed increased significantly. Steering responsiveness in a seaway was markedly reduced and it was difficult to hold a straight course. In the next series of tests, half the wedges were removed. The results were the same as in the first series, only less severe.

It is believed that the wedges induced a large drag which offset any possible savings due to trim changes. It would be preferable to induce trim changes by moving onboard weights. If a more fuel-efficient trim angle can be found, the steering problem can probably be corrected by increasing skeg area and moving the centroid of the underwater lateral plane further aft.

TABLE D-1

Effect of Fuel Load on Speed and Fuel Consumption

Engine RPM	Full Load (500 gal.) Average Fuel (GPH)	Full Load (500 gal.) Average Speed (KTS)	Half Load (250 gal.) Average Fuel (GPH)	Half Load (250 gal.) Average Speed (KTS)
600	1.55	5.34	1.55	5.33
1000	4.14	8.21	4.05	8.30
1500	11.24	10.48	11.14	10.76
2200	27.99	19.01	26.56	19.65
2400	33.09	22.30	31.33	22.43
2600	37.71	24.38	36.92	24.74

Notes:

1. Values are average of four runs.
2. Generally, the 41 ft. UTB goes about 0.4 KTS faster at half load while burning about 2% less fuel. See following graph.

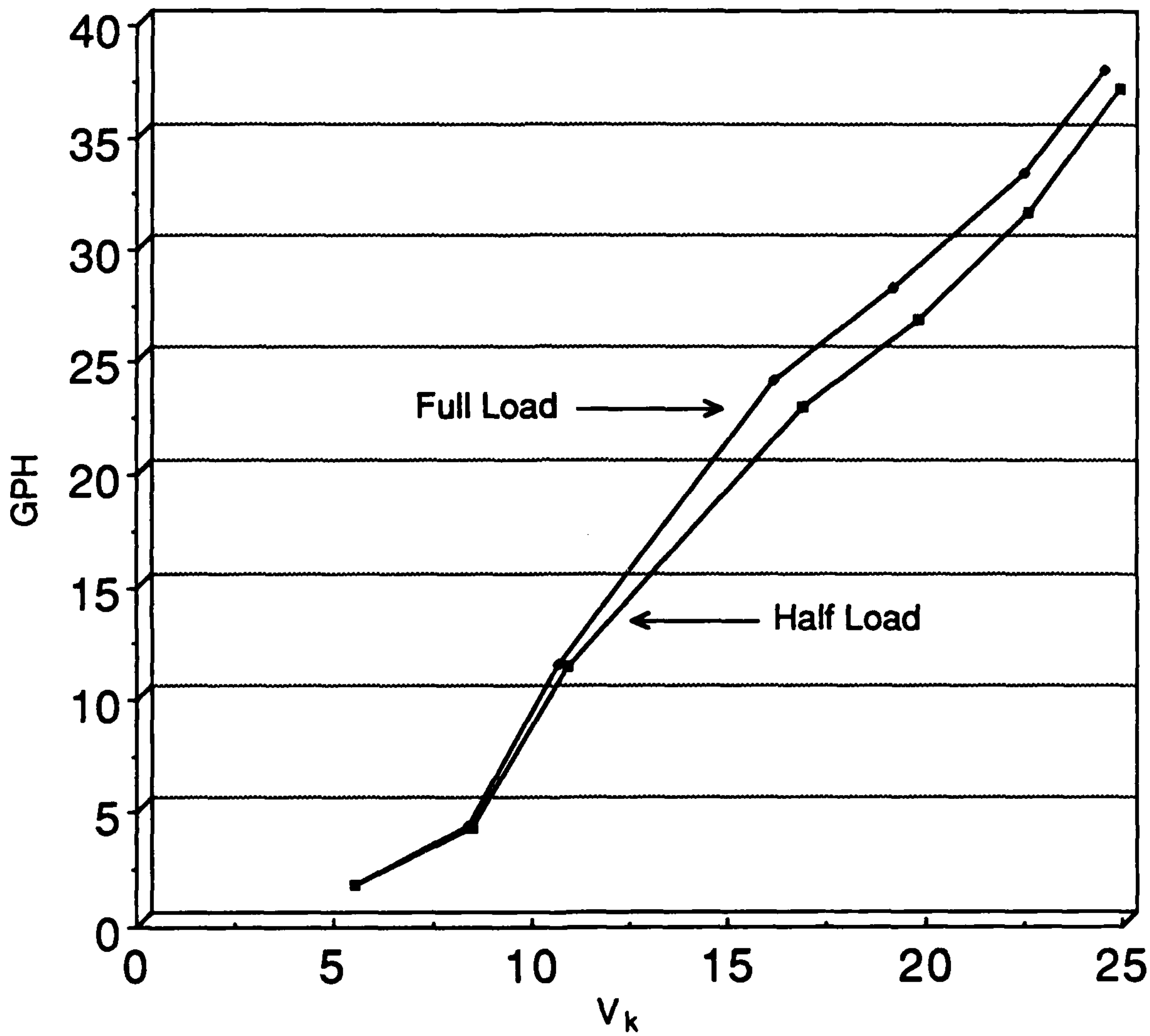

FIGURE D-1 Speed vs Fuel Rate at Full and Half Load